

Hybrid Liquid Desiccant Based Air-Conditioning Systems

by

C. S. Khalid Ahmed

A Thesis Presented to the

FACULTY OF THE COLLEGE OF GRADUATE STUDIES

KING FAHD UNIVERSITY OF PETROLEUM & MINERALS

DHAHRAN, SAUDI ARABIA

In Partial Fulfillment of the
Requirements for the Degree of

MASTER OF SCIENCE

In

MECHANICAL ENGINEERING

May, 1996

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Mechanical Engineering

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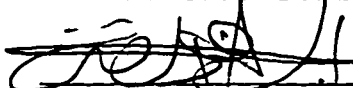
COLLEGE OF GRADUATE STUDIES

This thesis, written by **C. S. Khalid Ahmed** under the direction of his Thesis Advisor and approved by his Thesis Committee, has been presented to and accepted by the Dean of the College of Graduate Studies, in partial fulfillment of the requirements for the degree of **MASTER OF SCIENCE** in **MECHANICAL ENGINEERING**.

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Dedicated to

My

PARENTS,

SISTERS

and

BROTHERS

whose patience and perseverance

led me to this accomplishment.

Acknowledgments

First of all I thank Almighty **ALLAH (SWT)**, the only **LORD** and **CREATOR** for His Eternal Blessings and Guidance.

I would like to thank the King Fahd University of Petroleum and Minerals and particularly Dr. Muhammed O. Budair, chairman, Mechanical Engineering Department, for providing me all the facilities and requirements for achieving my masters degree. I am grateful to my advisor, Dr. P. Gandhidasan, for leading me in this area of research and for his consistent personal encouragement. I am thankful to my co-advisor, Dr. S. M. Zubair, for his valuable suggestions throughout my research work. I am sincerely thankful to my committee member, Dr. A. A. Al Farayedhi, for providing me all the necessary guidance for performing the experimental setup.

My appreciations are also due to Mr. Younus and Mr. Karam, fitting shop, who helped me lot during my experimental work. I acknowledge the timely help and friendly support given by my friends Moizuddin, Ilias, Ahmed Mohiuddin, Mohsin, Javeed, Shahid, Younus Ahmed, Syed Younus, Imtiaz, Firoz Daud, Firoz Ahmad, Nizam, Sultan Khan, and Abdul Razzack.

Finally I thank my Parents, Sisters, and Brothers for their constant advice and prayers.

Contents

Acknowledgements	i
List of Figures	vii
List of Tables	xi
Abstract (English)	xii
Abstract (Arabic)	xiii
Nomenclature	xiv
1 INTRODUCTION	1
1.1 Present Study	4
2 LITERATURE REVIEW	7
2.1 Hybrid Cycle Analysis	7
2.2 Open Cycle Regeneration	9

2.3	Combined Cycle Dehumidification Process	12
2.4	Economic Analysis of the Hybrid Cycle	13
2.5	Exergy Analysis	14
3	DESICCANT COOLING SYSTEMS	17
3.1	Desiccants	17
3.1.1	Desiccant Classification	18
3.2	Desiccant Cooling Potential	21
3.3	Desiccant Applications	23
3.4	The Desiccant Cycle	26
3.5	Desiccant Cooling System Components	30
3.6	Hybrid Systems	32
3.6.1	Hybrid Solid Desiccant Systems	33
3.6.2	Hybrid Liquid Desiccant Systems	35
4	HYBRID CYCLE ANALYSIS	38
4.1	Introduction	38
4.2	Comparison of Absorption and Vapor Compression Systems	39
4.3	Absorption Air-Conditioning System	43
4.3.1	The Absorption Process	44
4.3.2	The Absorber and Evaporator	45
4.3.3	Recovery of the Refrigerant	49

4.3.4	The Heat Exchanger and Cooling Water Circuits	50
4.3.5	Absorbent and Refrigerant Characteristics: Crystallization . .	51
4.4	The Hybrid Cycle	52
4.4.1	Mass and Energy Balance	55
4.4.2	Dehumidifier Analysis	64
5	EXERGY ANALYSIS	70
5.1	Introduction	70
5.2	Regenerator Analysis	71
5.2.1	Glazed Section	71
5.2.2	Open Section	73
5.3	Exergy Analysis	74
5.3.1	Glazed Section	75
5.3.2	Open Section	75
5.3.3	Evaporator	76
5.3.4	Absorber	76
5.3.5	Heat Exchanger 1	77
5.3.6	Heat Exchanger 2	82
5.3.7	Throttling valve	82
5.3.8	Dehumidifier	83
6	EXPERIMENTAL STUDY OF SOLAR REGENERATORS	85

6.1	Introduction	85
6.2	Types of Solar Regenerators	86
6.2.1	Fully Open Regenerator	87
6.2.2	Fully Closed Regenerator	87
6.2.3	Partly Closed-Open Regenerator	90
6.2.4	Alternately Closed-Open Regenerator	95
6.3	Performance Evaluation	95
6.4	Experimental Setup	97
7	RESULTS AND DISCUSSION	101
7.1	Hybrid Cycle Analysis	101
7.2	Exergy Analysis	113
7.3	Experimental Analysis of Solar Regenerators	126
8	CONCLUSIONS AND RECOMMENDATIONS	143
8.1	Conclusions	143
8.2	Recommendations	145
	Appendix	146
	Appendix I	146
	Appendix II	149

Bibliography	151
Vita	156

List of Figures

3.1	Various Types of Desiccant Cooling Systems.	22
3.2	Desiccant Water Vapor Pressure as a Function of Desiccant Moisture Content and Temperature [35].	27
3.3	The Desiccant Cycle [35].	28
4.1	Vapor Compression Refrigeration Cycle.	41
4.2	Vapor Absorption Refrigeration Cycle.	42
4.3	The Refrigerant pump and the Chilled Water Distribution Coil Added to the Evaporator.	47
4.4	Hybrid Liquid Desiccant Based Air-Conditioning Cycle.	53
4.5	Hybrid Cycle Components: Absorber and Evaporator.	58
4.6	Hybrid Cycle Component: Regenerator	59
4.7	Hybrid Cycle Component: Dehumidifier	60
4.8	Hybrid Cycle Component: Heat Exchanger 1	61
4.9	Hybrid Cycle Component: Heat Exchanger 2	62

4.10	Differential Element of the Dehumidifier.	65
4.11	Hybrid Cycle Processes Shown on a Psychrometric Chart.	69
5.1	Exergy Analysis : Partly Closed-Open Solar Regenerator.	72
5.2	Dehumidifier	78
5.3	Heat Exchanger 1	79
5.4	Heat Exchanger 2	80
5.5	Absorber and Evaporator	81
6.1	Fully Open Solar Regenerator.	88
6.2	Thermal Network for Fully Open Solar Regenerator.	89
6.3	Fully Closed Solar Regenerator.	91
6.4	Thermal Network for Fully Closed Solar Regenerator.	92
6.5	Partly Closed-Open Solar Regenerator.	93
6.6	Alternately Closed-Open Solar Regenerator.	94
6.7	Experimental Setup for Performance Evaluation of Open-Cycle Solar Regenerators.	98
7.1	Variation of COP with Regenerator Temperature for Different Refrig- erant Temperatures.	103
7.2	Variation of COP with Humidity Ratio for Different Absorber Tem- peratures.	104

7.3	Variation of COP with Regenerator Temperature for Different Absorber Temperatures	105
7.4	Variation of COP with Regenerator Temperature for Different Evaporator Temperatures.	106
7.5	Nomograph for Determining the Minimum Temperature Required in the Regenerator	107
7.6	Variation of Refrigerating Efficiency with Regenerator Temperature for Different Refrigerant Temperatures	108
7.7	Variation of Refrigerating Efficiency with Humidity Ratio for Different Absorber Temperatures.	109
7.8	Variation of Refrigerating Efficiency with Regenerator Temperature for Different Absorber Temperatures.	110
7.9	Variation of Refrigerating Efficiency with Regenerator Temperature for Different Evaporator Temperatures	111
7.10	Effect of Desiccant Mass Flow Rate on Irreversibilities as a Function of Ambient Temperature.	116
7.11	Effect of Insolation on Irreversibilities as a Function of Mass Flow Rate.	118
7.12	Effect of Desiccant Inlet Concentration on Irreversibilities as a Function of Mass Flow Rate.	119
7.13	Effect of Desiccant Inlet Temperature on Irreversibilities as a Function of Mass Flow Rate.	121

7.14 Effect of Ambient Vapor Pressure on Irreversibilities as a Function of Mass Flow Rate.	122
7.15 Effect of Ambient Temperature on Irreversibilities as a Function of Mass Flow Rate.	123
7.16 Effect of Ambient Wind Velocity on Irreversibilities as a Function of Mass Flow Rate.	124
7.17 Irrversible Losses in the Hybrid Cycle Components.	125
7.18 Temperature Profile in a Fully Open Solar Regenerator.	129
7.19 Moisture Evaporation Rate in a Fully Open Solar Regenerator.	130
7.20 Temperature Profile in a Fully Closed Solar Regenerator.	131
7.21 Moisture Evaporation Rate in a Fully Closed Solar Regenerator.	132
7.22 Moisture Evaporation as a Function of Glass Cover Height (Nov.1995).	133
7.23 Moisture Evaporation as a Function of Glass Cover Height (June 1996).	134
7.24 Moisture Evaporation Rate in a Partly Closed-Open Solar Regenerator.	136
7.25 Temperature Profile in a Partly Close-Open Solar Regenerator.	137
7.26 Moisture Evaporation Rate in an Alternately Closed-Open Solar Re- generator.	138
7.27 Temperature Profile in an Alternately Closed-Open Solar Regenerator.	139

List of Tables

4.1	Hybrid Cycle Parameters	63
5.1	Exergy Analysis:Component Parameters	84
6.1	Dates on which Experiments were Conducted	100
7.1	Regenerator Parameters	114
7.2	Experimental Results: Fully Open Solar Regenerator	141
7.3	Experimental Results: Fully Closed Solar Regenerator	141
7.4	Experimental Results: Partly Closed-Open Solar Regenerator	142
7.5	Experimental Results: Alternately Closed-Open Solar Regenerator . .	142

Abstract

Name: C. S. Khalid Ahmed

Title: Hybrid Liquid Desiccant Based Air-Conditioning System:

Major Field: Mechanical Engineering

Date of Degree: May 1996

A hybrid open-cycle vapor absorption (using LiBr as absorbent and H₂O as refrigerant) and liquid desiccant system has been simulated. LiBr has been used for the process of absorption in the absorber and dehumidification in the dehumidifier. The simulation model of the hybrid cycle is formulated with partly closed-open solar regenerator for regenerating the weak desiccant and a packed tower dehumidifier for the dehumidification of ambient air. The air is first dehumidified in the dehumidifier and then sensibly cooled in the evaporator before sending it into the comfort space. Subroutines for each component of the cycle have been used in order to calculate the varying parameters at all points in the cycle. An experimental setup using calcium chloride as the liquid desiccant has been carried out for the performance evaluation of four different types (fully open, fully closed, partly closed-open, and alternately closed-open) of solar regenerators suitable for regenerating the weak desiccant solution in the hybrid cycle. Exergy analysis for each component of the hybrid cycle has also been done to evaluate the irreversibilities in each of these components. Major emphasis has been given to the exergy analysis of the partly closed-open type of solar regenerator.

Master of Science Degree

King Fahd University of Petroleum and Minerals

Dhahran, Saudi Arabia

May, 1996

ملخص الرسالة

اسم الباحث : خالد أحمد
عنوان الرسالة : نظام تكييف هواء مهجن من دوائر التبريد بالإمتصاص والسوائل المجففة

التخصص الرئيسي : الهندسة الميكانيكية
تاريخ الدرجة : مايو ١٩٩٦

في هذا البحث تمت محاكاة (باستخدام الحاسب الآلي) دورة مفتوحة مهجنة من نظامين الأول دورة تبريد بامتصاص البخار (تستخدم محلول الليثيوم بروميركوسط امتصاص والمياه كوسيط تبريد) والثاني هو السوائل المجففة محلول الليثيوم بروميد قد استخدم لعملية الإمتصاص وإزالة الرطوبة . وقد تم بناء نموذج محاكاة هذه الدورة المهجنة ذات المسترجع الشمسي المفتوح - المغلق جزئيا لاسترجاع المجفف الضعيف وأبراج معبأة لانتزاع رطوبة الهواء الجوي . أولا تنزع رطوبة الهواء في هذه الأبراج ثم يبرد حراريا فقط في المبخر قبل إرساله الى المكان المكيف . ولقد تم استخدام برامج فرعية لكل جزء في الدورة لحساب المعاملات المتغيرة عند كل نقطة في الدورة . كما تم انشاء جهاز تجريبي يستخدم محلول الكالسيوم كلوريد كسائل تجفيف وذلك لتقييم اداء اربعة انواع مختلفة (نوع مفتوح تماما ، مغلق تماما ، مفتوح - مغلق جزئيا ، مفتوح - مغلق تبادليا) للمسترجع الشمسي المناسب لاسترجاع سائل التجفيف الضعيف في هذه الدورة المهجنة . كما تم ايضا اجراء تحليل الأكسرجي في كل جزء من الدورة المهجنة وذلك لتقييم الانعكاسية في كل من هذه الأجزاء مع اعطاء تركيز رئيسي لتحليل الأكسرجي للمسترجع الشمسي المفتوح - مغلق جزئيا .

درجة الماجستير في العلوم
جامعة الملك فهد للبترول والمعادن
الظهران ، المملكة العربية السعودية
مايو ١٩٩٦ م

Nomenclature

A	surface area, m^2
a	specific interfacial surface, m^2/m^3
a, b, c	empirical constants
C	specific heat, kJ/kg K
COP	coefficient of performance
e	exergy, kJ/h
F	mass transfer coefficient, $\text{kmol/m}^2\text{s}$
GM	molar gas mass velocity at inlet, $\text{kmol/m}^2 \text{ s}$
g	heat transfer flux of gas, $\text{kJ/m}^2\text{s}$
G	mass flow rate of desiccant, kg/h m
h_{Ga}	volumetric heat transfer coefficient of gas phase, $\text{W/m}^3 \text{ K}$
h'_{Ga}	volumetric heat transfer coefficient of gas phase corrected for mass transfer, $\text{W/m}^3\text{K}$
H	molar enthalpy of gas stream, kJ/kmol
h	enthalpy, kJ/kg
h_{fg}	latent heat of vaporisation, kJ/kg
I	insolation, kJ/ h m^2

\dot{I}	irreversibilities, kW
L	molar liquid mass velocity, kmol/m ² s
m	mass flow rate, kg/s
M	mass of water evaporated from the desiccant, kg /h m
MW	molecular weight, kg/kmol
N_C	molar flux of water relative to a fixed surface, kmol/m ² s
P	vapor pressure of water, mm Hg
Q	heat transfer rate, kW
\bar{R}	Universal gas constant, kJ/kgmole K
s	entropy, kJ/kg K
T	temperature, K
t	temperature, ° C
t_O	reference temperature, K
U_L	overall heat-loss coefficient, W/m ² –° C
V_e	velocity, m/s
X	concentration of LiBr, kg/kg of solution
x	mole fraction of water vapor
x_c	concentration of water in liquid phase, kmol H ₂ O/kmol liquid mixture
x_{cl}	value of x_c at interface, kmol H ₂ O/kmol liquid mixture
y_c	concentration of water in gas phase, kmol H ₂ O/kmol gas mixture

y_{cl}	value of y_c at interface, kmol H ₂ O/kmol gas mixture
Y	concentration of water (humidity ratio) in gas phase, kmol H ₂ O/kmol dry air
Z	height of the packing from the bottom, m

Greek symbols

α	absorptance of the liquid layer and the absorber surface
$(\alpha\tau)$	effective absorptance-transmittance product
β	mass transfer coefficient between solution film and air stream, kg/h m ² (mm Hg)
ΔH_S	integral heat of solution at the reference temperature, kJ/kmol
λ	molar latent heat of vaporization of water at the reference temperature, kJ/kmol
η	efficiency of the regenerator
η_{refg}	refrigerating efficiency
ρ	reflectance of the desiccant for solar radiation
ξ	concentration of the desiccant, kg/kg of solution

Subscripts

1, 2, 3, .. states points

ab	absorber
B	dry air
b	beginning of the regenerator
C	water vapor
ev	evaporator
G	gas stream
g	glazed section of the regenerator
gen	generator (regenerator)
gl	generator load
I	interface
i	inlet
L	liquid stream
l	total length of the partly closed-open regenerator
m	moisture
o	open section of the regenerator, outler
r	refrigerant
rd	amount on moisture absorbed in the dehumidifier
s	desiccant
st	total solution reaching the regenerator
t	total refrigerant

v	vapor
∞	ambient conditions

Chapter 1

INTRODUCTION

In recent years, ozone layer depletion and the greenhouse effect have created a considerable public concern. The fully halogenated chlorofluorocarbons (CFCs) which have provided much needed refrigeration and air-conditioning for about sixty years, are found to be among the gases responsible for the depletion of ozone layer, and for creating global warming. According to the Montreal protocol, these CFCs must be phased out by the year 2000 [1]. CFCs have an average lifetime of one hundred years in the atmosphere and today's emissions will continue to affect the ozone layer for another century or so. If CFC emissions were to grow 3% per year for the foreseeable future, predicted global ozone depletion would exceed 10% before the year 2050 [2]. Also, 65 percent of the CFCs produced annually are being used only to replace the leaked out CFCs and 35 percent is being used in manufacturing the new refrigerators and air-conditioners. The air-conditioning industry has now phased out the use of

R11 and R12 and has moved over to the use of R22 as a short-term solution [3]. A new HCFC (halogenated chlorofluorocarbons) refrigerant R123, which was expected to replace R11 in centrifugal chillers, has recently been discovered to be a carcinogen when present in atmosphere above certain limits, causing extreme doubts about its future use [2]. The refrigerant R134a, tetrafluoroethane, CH_2FCF_3 , contains no chlorine atoms and hence has a zero ozone depletion potential. Suitable lubricants have been found for R134a and at present it appears to be the best replacement refrigerant for CFCs and in the long term, HCFCs. The search for refrigerants and the adaptation of the existing machinery to the alternative refrigerants have created considerable research activities. Efforts have also been devoted to the developments of alternate methods of refrigeration and air-conditioning.

Solar energy is one of the alternative sources of energy which is free and provides no environmental pollution. The availability of this energy makes it highly attractive for use in space cooling applications. In an attempt to find a suitable solar system to produce cooling economically, a number of different approaches have been pursued by researchers. A great majority of these have dealt with the adaptation of absorption cooling systems and dehumidification processes. Although technically these approaches were found feasible, economically they were not viable due, mainly, to initial costs and the consumption of large quantities of electricity. Conventional vapor compression systems which operate of closed-cycle using electricity have reached their limits after 60 years of research and development. The cost of electricity in-

creases at a rate depending on the supply and demand. New cooling systems using renewable energy sources directly are being developed when the economics show a favorable rate of return. These energy sources impose constraints on the candidate systems.

Closed-cycle cooling systems in which the refrigerant does not contact the atmosphere cannot use the energy of the unsaturated atmosphere, since the working liquid (refrigerant) is not exposed to the atmosphere. Open-cycle systems evaporating the water into the atmosphere as a refrigerant use this energy source and use less direct solar energy; e.g., the simple evaporative cooler requires energy only to move the water and air. Closed systems may be combined with open systems to take advantage of this. The performance of open systems is reduced in humid areas unless dehumidification is used. A solar collector will deliver more heat at low temperatures because heat losses are less. These facts lead to choose open cycles using dehumidification with low temperature heat as promising options for a renewable energy cooling system.

Combination of a conventional air-conditioning system with a desiccant system to reduce the sensible load and latent load results in a hybrid system which controls both the temperature and the humidity ratio within the comfort range.

1.1 Present Study

The proposed research consists of a hybrid liquid desiccant system using $LiBr$ as the liquid desiccant for both absorption and dehumidification processes and water as refrigerant. The system operates on an open cycle in which the weak absorbent solution is regenerated by losing the refrigerant to the earth's atmosphere instead to the condenser as in a closed cycle system. Cooling takes place by evaporating refrigerant from an external source in the evaporator rather than obtaining refrigerant from the condenser. The weak absorbent solution is then pumped to the regenerator which is partly closed-open to the atmosphere through the regenerative heat exchanger. It is then heated and subsequently concentrated in the solar regenerator. The strong solution then passes to the absorber. In the absorber, the strong desiccant absorbs water vapor as soon as the incoming refrigerant evaporates in the evaporator. The heat of absorption for the refrigerant-absorber solution is removed by cooling tower loop. In the evaporator, water from an external source is evaporated at reduced pressure by the energy supplied by the heat from the cooled space. The resultant weak desiccant is then pumped from the absorber to the regenerator, completing the absorption-machine cycle. Part of the strong desiccant from the regenerator is passed to the dehumidifier, where the air is dehumidified in the counter-current direction. The actual dehumidification takes place due to the vapor pressure difference between the vapor in the air and the liquid desiccant. As the regenerated

liquid desiccant is cooled and concentrated, its vapor pressure is less than the vapor pressure of the air, therefore the vapor present in the air tends to escape into the liquid desiccant, thereby diluting the liquid desiccant. The weak desiccant from the dehumidifier is then pumped back for regeneration along with the weak desiccant from the absorber. After regeneration the strong solution is supplied to absorber and the dehumidifier in the required proportions.

The primary objectives of the present study are to,

1. perform mass and energy balance analysis for the hybrid vapor absorption and liquid desiccant cooling cycle.
2. study experimentally, the rate of moisture evaporation in a fully open, fully closed, partly closed-open, and alternately open and closed solar regenerators.
3. perform exergy analysis for the hybrid liquid desiccant air-conditioning cycle components.

To meet the above mentioned objectives a thorough literature review has been made and is discussed in chapter 2. The various desiccant cooling systems and its applications are explained in chapter 3. This is followed by a detail mass and energy analysis of the hybrid cycle in chapter 4. An exergy analysis for different components of the hybrid cycle is presented in chapter 5 which is followed by the experimental analysis for the open cycle regenerators in chapter 6. Results and discussion for the hybrid cycle analysis, exergy analysis, and the experimental work are presented

separately in chapter 7. Finally, conclusions and recommendations for the future work are presented.

Chapter 2

LITERATURE REVIEW

2.1 Hybrid Cycle Analysis

A combined absorption-desiccant sun powered air conditioning system was studied by Aly [4]. Silica gel beds of cross flow type was used as a desiccant for the process of dehumidification. The combined system COP achieved was 1.55, which is about 89% more than the absorption solar powered cycle. Similar integration of absorption-desiccant air conditioning system was done by Gari et al [5], but by using liquid desiccant. A case study was done by combining the $LiBr-H_2O$ absorption cycle with a liquid desiccant cycle. An overall COP of 1.21 was obtained which is about 50% higher than that expected from the absorption cooling machine alone. Khamid and Proselkov [6], presented a thermodynamic analysis and choice of design parameters of solar absorption lithium bromide refrigerating machines. A group of formulas

to determine the basic thermodynamic properties of an LiBr-H₂O solution were presented. The effect of different thermodynamic parameters and temperatures on the COP of the absorption refrigerating machine were also presented.

A detailed review of desiccant cooling systems was done by Waugaman et al [7]. The desiccant cooling cycles and their application to air-conditioning systems has been described briefly. The possible combination of desiccant dehumidification with conventional air-conditioning systems have been presented by reviewing all the latest research works. It has also been described that desiccant cooling systems are being found their places in residential and commercial applications inspite of their initial costs being comparatively high. A solar assisted open cycle absorption cooling system was demonstrated by Hawlader [1]. A review of the research activities performed in the area of solar assisted open-cycle absorption cooling was presented. The various variables affecting the performance of the open -cycle has also been studied.

An experimental investigation of a combined solar assisted desiccant-evaporative air conditioning system was done by Khalid and Shakir [8]. The solar air-heater regeneration temperature was constant at 70°C and the solid desiccant used was a rotary disc of silica gel. A maximum seasonal COP of 2.8 was obtained for a mass flow rate of 0.075 kg/s. The performance of the system was found to be improved with higher regeneration temperature, higher process air mass flow rate, and dry weather. A similar performance analysis of an open-cycle liquid desiccant cooling

system using solar energy for regeneration was done by Gandhidasan [9]. The system operates on ventilation mode with evaporative cooling and has been proposed for high humidity and temperature regions like Dhahran, Saudi Arabia. In this system, the ambient air is dehumidified using a liquid desiccant followed by adiabatic cooling and the weak desiccant is regenerated by solar energy. The effects of ambient air conditions, insolation, cooling water temperature, and the desiccant concentration on the performance of the dehumidifier have also been demonstrated. Jain et al [10], evaluated liquid desiccant based evaporative cooling cycles for tropical hot and humid climates. A psychometric study for the various liquid desiccants was performed for 16 tropical Indian cities. The effect of specific humidity, outlet air temperature, and volumetric air flow on the COP of the cycle was also studied. It was shown that the combination of liquid desiccant dehumidifier and the evaporative cooler is better in terms of COP for a wide range of outdoor conditions. The results presented in the article are useful in the design of liquid desiccant based air-conditioning systems suitable for the monsoon season in tropical countries like India.

2.2 Open Cycle Regeneration

In 1942, Berestneff [11] described the process of dehumidification of humid air due to the vapor pressure difference of moisture in the air and at the medium of ab-

sorption in an open absorption system. The process of absorption of moisture by the medium of absorption depends on its concentration, temperature and the vapor pressure. The higher the concentration of the solution at a certain temperature, lower the partial pressure, and higher the temperature of the solution at certain concentration, higher the partial vapor pressure of moisture in it. Therefore, the partial pressure of the solution should be low in order to prevent its escaping into the air. The medium of absorption used in this study was Li Cl. Kakabaev and Khandurdyev [12] demonstrated an absorption solar refrigeration unit with open regeneration of the solution by utilizing solar energy in regions of large amounts of sunshine and dry climates. An expression for mass of evaporation of moisture from the regenerator was derived and the various factors which affect the evaporation rate was studied. An empirical relation between vapor pressure, temperature, and concentration of the desiccant for LiCl and LiBr was also presented. Khallyev [13] demonstrated the effect of high outside humidity on the efficiency of solar regenerator. The regenerator considered in this study was partly closed-open, with the glazed portion at the beginning of the regenerator. The glazed section was used to increase the temperature of the incoming weak solution and the open section was used to evaporate the moisture from the weak solution.

Gandhidasan and Farayedhi [14] studied the effect of insolation, solution concentration, solution flow rate, ambient water vapor pressure and temperature on the efficiency of the partly closed-open solar regenerator. Their study concluded that the

efficiency of the regenerator decreased with the increase in the ambient water vapor pressure, and the maximum efficiency could be reached at a flow rate of 5 kg/h m^2 of the solution. An analysis and simulation of an open absorption refrigeration system was done by Collier [15]. The relationship between the collector length and the solution mass flow rate was tied to the environmental factors such as wind velocity and humidity. In this study, the overall daily cooling COPs ranged from 0.09 to 0.45 for various conditions. Theoretical studies on solar collectors for open cycle absorption air-conditioning using liquid desiccants were done by Kaushik et al [16]. This study concluded that the collectors performed better for high solar insolation and ambient air temperature and low solution concentration with atmospheric water vapor pressure in the medium range. A similar theoretical study (based on enthalpy and mass balance) for the regeneration of liquid desiccants was done by Fountoukipis et al [17]. A comparison between the theoretical predictions and experimental values was found to be in total agreement with each other. The model developed can be used to predict the regeneration of liquid desiccants if the air temperature, humidity, the air velocity, and the solar insolation are known.

Performance evaluation of glazed and unglazed solar collectors was done by Hawlader et al [18]. It was found that the performance of unglazed solar regenerator was better than that of a glazed regenerator. This performance was evaluated in terms of quantity of moisture evaporated from the regenerator. It was found that the performance of the glazed regenerator approaches that of the unglazed regenerator

by increasing the height of glazing. A report on experimental study of the heat and mass transfer in solar collector/regenerator for open-cycle liquid desiccant regeneration was prepared by Ji and Wood [19]. This report gives an account of research conducted in the systems analysis and applications research activity area concerning open-cycle solar cooling concepts. This report also consists of experimental work to identify and prioritize the major technical issues to advance the potentially lower cost of open-cycle solar regeneration.

2.3 Combined Cycle Dehumidification Process

A packed bed dehumidifier/regenerator for solar air conditioning with liquid desiccants was proposed by Factor and Grossman [20]. An experimental apparatus was constructed and experiments were performed with LiBr and Monoethylene glycol as desiccants. LiBr experiments showed a very good agreement with theoretical model. It was found that pressure drop can be reduced with suitable packing material in the dehumidifier. Ullah et al [21] demonstrated the effectiveness of moisture removal of an adiabatic packed tower dehumidifier operating with $CaCl_2$ - air contact system. Queiroz et al. [22] developed a model to analyse the performance of a non-adiabatic liquid dehumidifier used in solar air conditioning systems. In this setup, air and tri-ethylene glycol circulate counter currently outside staggered copper tubes which are the filling of the dehumidifier tower. The mass transfer coefficient calculations

are based on merkel integral approach. Calculation of heat and mass transfer coefficients in a packed tower operating with a desiccant-air contact system was presented by Gandhidasan et al. [23]. In this study, aqueous calcium chloride solution was used. Ceramic raschig rings and Berl saddles were used as the packing material in this study. The air phases transfer coefficients are correlated with flow rates of air and liquid and the temperature of air, whereas liquid phase coefficients are correlated with rates of air and liquid flow, and the temperature and concentration of the liquid.

2.4 Economic Analysis of the Hybrid Cycle

A technoeconomic assessment and parametric study of vapor-compression and solid/liquid desiccant hybrid solar space conditioning system was studied by Yadav and Kaushik [24]. A hybrid system with desiccant cycle is found to be more promising under high latent heat load and higher ambient humidity conditions and significant energy savings can be achieved over standard vapor compression cooling systems. Jurinak et al. [25] have discussed as to how open-cycle desiccant air conditioning system can be an alternative to vapor compression cooling in residential applications. The comparison is done on the basis of primary energy use and cost. Desiccant systems with improved dehumidifiers are found to achieve seasonal COPs of about 1.1.

A similar comparison analysis between liquid desiccant cooling systems and con-

ventional chilled water cooling system was done by Sick et al. [26]. The comparison is done on the basis of seasonal thermal and electrical energy use and operational costs. It is found that the cooling energy required by a conventional conditioner exceeds the cooling requirements of the load. Therefore, a preferred liquid desiccant configuration from an operating cost standpoint is proposed. Warren and Wahlig [27] gave a model description and simulation results for the analysis and comparison of active solar solid desiccant and absorption cooling systems. This comparison was done between ventilation mode desiccant system, advanced absorption systems and conventional vapor compression systems. The economic analysis has been done to study each component individually before comparing them with different cooling systems. Using a medium-sized office building with a variable air volume systems as a test case, electrical, thermal and overall system performance at ARI outdoor conditions have been calculated for vapor compression, absorption and desiccant cooling systems.

2.5 Exergy Analysis

Fujiwara [28], discussed the exergy analysis for the performance evaluation of solar collectors. Optimum operating conditions of the solar collector were presented considering the pressure drop and pressure losses inside the collectors. The maximum exergy of the collectors was expressed in terms of the collector parameters and

the environment in which they operate. The thermal efficiency of the collector was compared with the exergy efficiency for different operating conditions.

A second law analysis and synthesis of solar collector systems considering the effects of heat transfer irreversibilities occurring between the sun and the solar collector, between the collector and the ambient air, and inside the collector was presented by Bejan et al. [29]. The optimum operating conditions for minimum heat transfer irreversibilities were derived for isothermal collector, non isothermal collector, and collector-user heat exchanger system. Different non dimensional parameters were derived to reduce the complication of understanding the second law analysis of the solar collectors. Bejan [30] considered the effect of time varying conditions on the exergy extraction rate of the solar collectors. The exergy storage and its usage was presented by constructing a solar collector model with a storage system. It was shown that the practice of operating collectors at constant temperature, regardless of time of the day, is responsible for a sizeable and steady exergy loss. The maximum exergy that can be extracted from the collector with thermal inertia was also expressed. Performance of two flat-plate thermal solar collectors and one photovoltaic panel was analyzed based on both first and second laws of thermodynamics by Said and Zubair [31]. The analysis presented was based on measurements that were obtained from an experimental solar test facility.

Chelghoum and Bejan [32] discussed the second law analysis of solar collectors with energy storage capability. An analytical as well as numerical study was done for

the exergy that can be delivered by solar collector installation with temporary energy storage capability. It was analysed that the exergy delivered was mostly affected by the relative timing of the filling and discharge processes. The daily regime of operation of the collection/ storage installation was also considered in the design in order to maximize the harvesting of solar exergy per unit collector area. A second law analysis of absorption cooling cycles was discussed by Egrican [33]. The second law of thermodynamics was applied to various components of the absorption cycle. Two new effectiveness factors were defined for two working fluids and the choice of the working fluids based on second law of thermodynamics was presented. Lithium bromide / water was found to be an alternative absorbent/cooling combination which does not destroy ozone layer.

Aphornratana and Eames [34] made a thermodynamic analysis of absorption refrigeration cycles using the second law of thermodynamics method. The second law analysis was applied to a single effect absorption refrigeration cycle. A novel method for calculating the entropy of lithium bromide and water solution was also presented. Exergy change in each component of the absorption machine as a percentage of the exergy input from the generator was analysed.

Chapter 3

DESICCANT COOLING SYSTEMS

3.1 Desiccants

Materials which have the tendency to attract and hold the water vapor are termed as desiccants. A commercial desiccant takes up between 10 and 1100% of its dry weight in water vapor, depending on its type and the moisture available in the environment [35]. Further commercial desiccants continue to attract moisture even when the surrounding air is quite dry, a characteristic which other materials do not share. All desiccants behave in a similar manner i.e., they attract moisture until they reach equilibrium with the surrounding air. Moisture is usually removed from the desiccant by heating it to temperatures between 50 and 250⁰ C and exposing it to a scavenger

airstream. After the desiccant dries it must be cooled so that it can attract moisture once again. Sorption (absorption in case of liquid desiccants and adsorption in case of solid desiccants) always generates sensible heat equal to the latent heat of the water vapor taken up by the desiccant, plus an additional heat of sorption that carries between 5 and 25% of the latent heat of the water vapor. This heat is transferred to the desiccant and the surrounding air. The process of attracting and holding moisture is described as absorption or adsorption, depending whether the desiccant undergoes a chemical change as it takes on moisture. Adsorption does change the desiccant except by the addition of the weight of water vapor. Absorption, on the other hand, changes from a solid to a liquid as it absorbs moisture.

3.1.1 Desiccant Classification

Desiccants can be solids or liquids and can hold moisture through adsorption or absorption. Most absorbents are liquids, and most adsorbents are solids. Adsorbents are solid material with a tremendous internal surface area per unit of mass, a single gram can have more than 4600 m² of surface area [35]. The bulk of the adsorbed water is contained by condensation into the capillaries, and the majority of the surface area that attracts individual water molecules is in the crystalline structure of the material itself. Adsorbents attract moisture because of the electric field at the desiccant surface. The field is not uniform in either force or charge, so it attracts water molecules that have a net opposite charge from specific sites on the desiccant

surface. When the complete surface is covered, the adsorbent can hold still more moisture as vapor condenses into the first layer and fills the capillaries throughout the material.

General classes of solid desiccants are,

1. Silica Gels
2. Zeolites
3. Activated Aluminas
4. Carbons

Liquid desiccant dehumidification can best be illustrated by comparing it to the operation of an air washer. When air passes through an air washer, its dew point approaches that of the temperature of the water supplied to the machine. More humid air is dehumidified and less humid air is humidified. In a similar manner, a liquid desiccant dehumidifier contacts air with a liquid desiccant solution. The liquid has a vapor pressure lower than water at the same temperature, and the air passing over the solution approaches this reduced vapor pressure; it is dehumidified. The vapor pressure of a liquid desiccant solution is directly proportional to its temperature and inversely proportional to its concentration. As the desiccant content in the mixture increases, its vapor pressure decreases. This pressure difference allows the desiccant solution to absorb moisture from the air whenever the vapor pressure of the air is greater than that of the solution.

In standard practice, the behavior of a liquid desiccant can be controlled by adjusting its concentration, its temperature, or both. Concentration is controlled by heating the desiccant to drive moisture out into a waste airstream or directly to the ambient. Commercially available liquid desiccants have an especially high water holding capacity. Each molecule of lithium chloride, for example, can hold two water molecules even in dry state. Above two water molecules per molecule of LiCl, the desiccant becomes a liquid and continues to absorb moisture. If the solution is in equilibrium with air at a 90% relative humidity condition, approximately 26 water molecules are attached to each molecule of LiCl. This represents a water absorption of more than 1000% on a dry weight basis [35]. As a practical matter, however, the absorption process is limited by the surface area of a desiccant exposed to the air being dehumidified and the contact time allowed for the reaction. More surface area and more contact time allows the desiccant to approach its theoretical capacity. Commercial desiccant systems reflect these realities either by spraying the desiccant onto an extended surface much like in a cooling tower, or holding a solution in a rotating extended surface with a large solution capacity. In the present research work, liquid desiccants have been chosen for the process of absorption and dehumidification.

3.2 Desiccant Cooling Potential

The various types of desiccant cooling systems are shown in Fig.3.1. Both solid and liquid desiccants are used in open-cycle cooling systems. A solid desiccant is used in closed-cycle cooling systems. The present study deals with the Integrated systems which consists of combining either a vapor compression or vapor absorption machine with a liquid or solid desiccant systems.

The advantages of desiccant cooling systems are:

1. They use no CFCs.
2. They improve indoor air quality.
3. They conserve energy and can be operated using renewable forms of energy (solar or waste heat).
4. The market penetration of desiccant systems is 30% for residential buildings and 20% for commercial buildings, i.e., either 30% or 20% of the existing conventional systems will be replaced with desiccant systems. For commercial applications the desiccant system will be probably used for latent load removal and will be integrated with the existing HVAC systems [36].
5. The thermal coefficient of performances (COPs) of the desiccant systems are found to be around 1.5. There is a prediction that with the recent and expected advances in the desiccant technology further higher COPs can be attained [36].

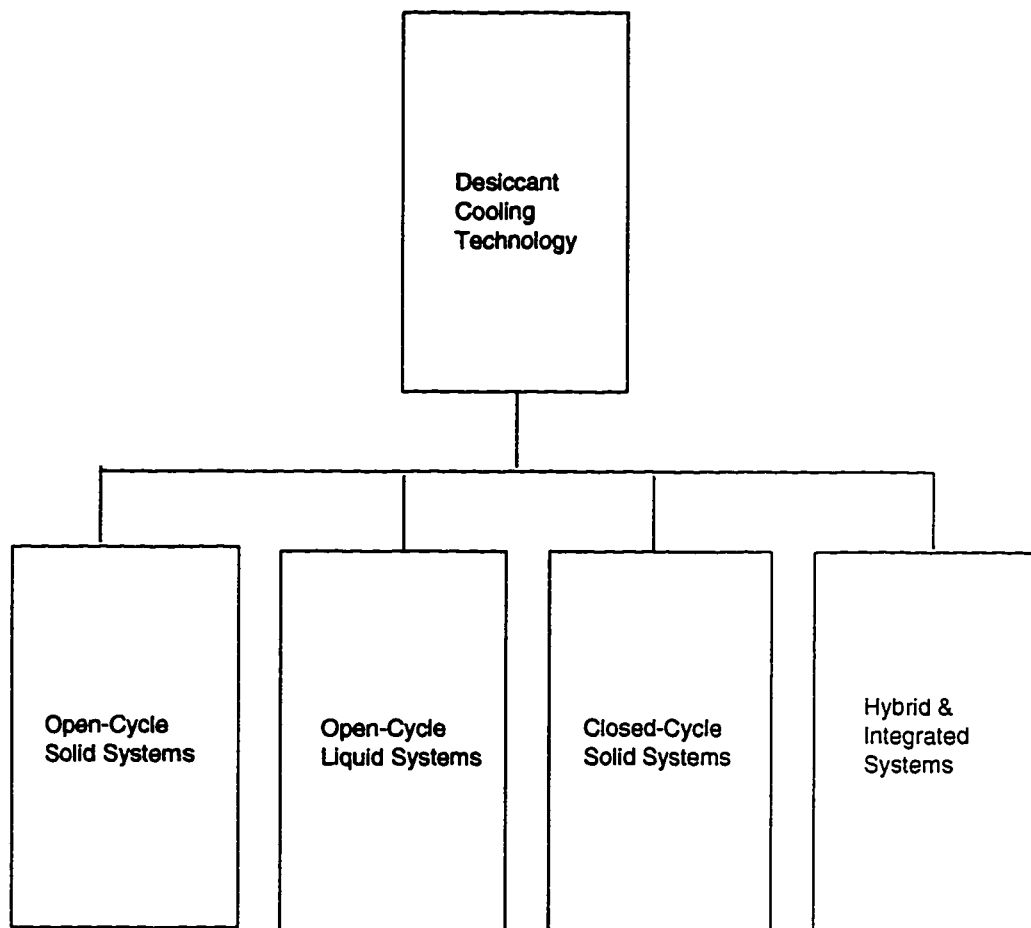


Figure 3.1: Various Types of Desiccant Cooling Systems.

Conventional vapor compression / absorption systems are preferred for providing sensible cooling. To generate a latent cooling capacity, however, they must cool the air passed to the dew point to condense and remove the excess moisture. Desiccant dehumidifier cooling systems provide latent cooling, but must overdry the air, so sufficient sensible and latent cooling capacity remains after the evaporative cooling process.

3.3 Desiccant Applications

Desiccants can dry either liquids or gases, including ambient air, and are used in many air-conditioning applications particularly when [35],

1. The latent load is large in comparison to the sensible load.
2. The cost of energy to regenerate the desiccant is low when compared to the cost of energy to dehumidify the air by chilling it below its dew point.
3. The moisture control level required in the control space would require chilling the air to subfreezing dew points if compression refrigeration alone were used to dehumidify the air .
4. The temperature control level required by the space or process requires continuous delivery of air at subfreezing temperatures.

In many of these situations, the cost of running a vapor compression cooling system can be very high. A desiccant process may offer considerable advantages in energy, the initial cost of the equipment, and maintenance. Since desiccants are able to absorb more than simply water vapor, they can remove contaminants from airstreams to improve indoor air quality. Desiccants have been used to remove organic vapors and in special circumstances to control microbiological contaminants. Desiccants are also used in drying compressed air to low dew points. In this application, moisture can be removed from desiccant without heat. Desorption is accomplished using differences in vapor pressures compared to the total pressures of the compressed and ambient pressure airstreams. Desiccants are also used to dry the refrigerant circulating in air-conditioning and refrigeration systems. This reduces corrosion in refrigerant piping and avoids valves and capillaries becoming clogged with ice crystals. In this application the desiccant is not regenerated; it is discarded when it has adsorbed its limits of water vapor. One of the functions of a building is to maintain comfortable temperatures and humidity. Reducing the non-renewable energy required to achieve this is a major goal. In summer, infiltration, ventilation, solar gains, building envelope condition, internal heat and moisture generated by the occupants, and equipment raise the temperature and humidity of the conditioned space. Better equipment, building design, and insulation, sealing, and ventilation systems can reduce but not eliminate five of these heat loads. The heat and moisture generated by the occupants depend on building function and cannot

be significantly altered. A cooling system with reduced energy consumption, especially one using renewable energy that can be nationally marketed and retrofitted to existing building stock, is an extremely attractive alternative to the consumer utility and the industry that supplies and installs the product [36].

A properly marketed and retrofitted system ideally needs to be compact and have low cost and maintenance with high performance and reliability. High performance can be achieved as the system approaches the limits set by the second law of thermodynamics. This requires that all heat transfer should be across small temperature differences and all mass transfers across small concentration differences. Of commercial dehumidifiers only the adiabatic regenerative dehumidifier does this. The design of regenerative dehumidifiers is complicated by mass transfer resistances and poor manufacturing and mechanical properties of available desiccants. New materials, geometries, and manufacturing techniques are capable of reducing size, cost, and maintenance while increasing the performance and reliability of the regenerative dehumidifiers. An improved adiabatic regenerative dehumidifier will facilitate national marketing and retrofitting of desiccant open cycle cooling systems. The open cycle absorption cooling system, where a liquid absorbent is exposed to the atmosphere through a glazed /unglazed /partly glazed-open solar collector for regeneration is another approach which has stimulated considerable interest. This system uses solar energy to increase the concentration of a desiccant, acting as the absorbent, by evaporating a portion of the refrigerant (water). The increased concentration leads

to higher chemical potential, which enables it to absorb the vapor in the absorber and drive the cooling process by continuing the cycle.

3.4 The Desiccant Cycle

Practically speaking, all desiccants function by the same mechanism- transferring moisture because of difference between the water vapor pressure at their surface and that of the surrounding air. When the vapor pressure at desiccant surface is lower than that of air, the desiccant attracts moisture. When the surface vapor pressure is higher than that of the surrounding air, the desiccant releases moisture.

Figure 3.2 shows the relationship between the moisture content of the desiccant and its surface vapor pressure as a function of temperature. As the moisture content of the desiccant rises, so does the the water vapor pressure at its surface. At some point, the vapor pressure at the desiccant is the same as that of the air - the two are in equilibrium. Then moisture cannot move in either direction until some external force changes the vapor pressure at the desiccant or in the air. Higher temperatures and moisture content increases the vapor pressure at the surface. When the surface vapor pressure exceeds that of the surrounding air, moisture leaves the desiccant- a process called reactivation or regeneration. After the desiccant is dried by the heat, its vapor pressure remains high, so that it has very little ability to absorb moisture. Cooling the desiccant reduces its surface vapor pressure so it can absorb moisture

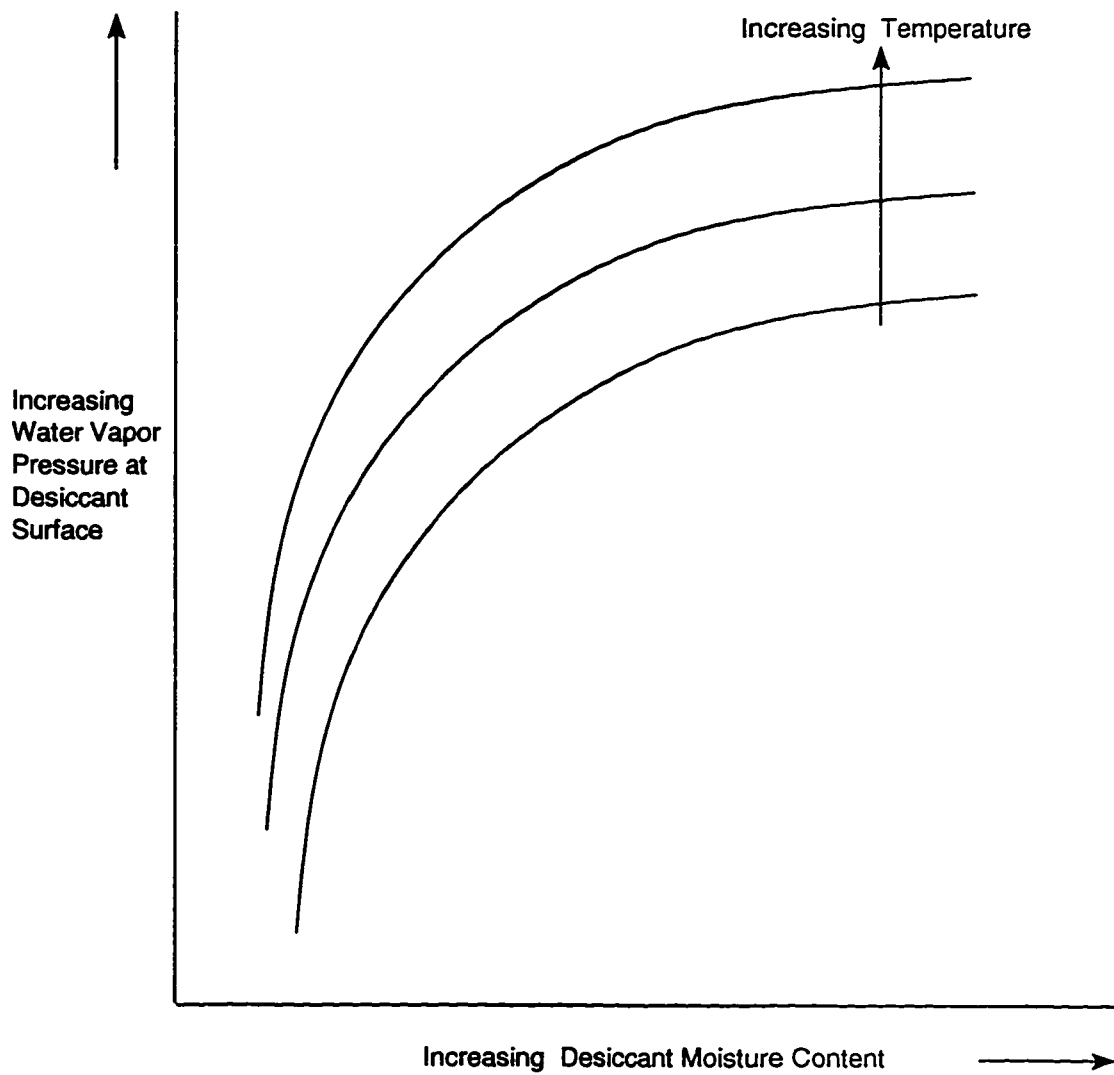


Figure 3.2: Desiccant Water Vapor Pressure as a Function of Desiccant Moisture Content and Temperature [35].

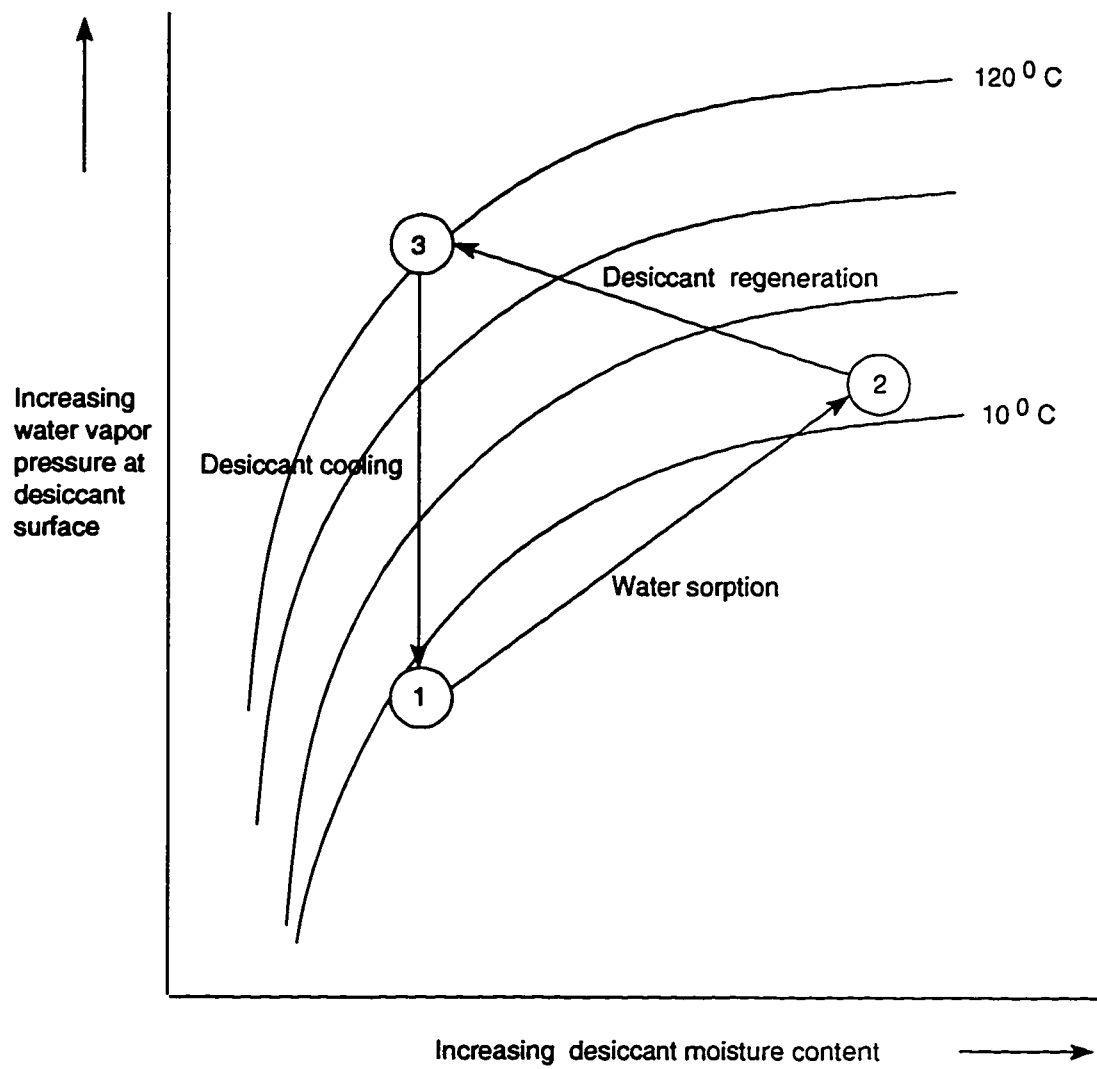


Figure 3.3: The Desiccant Cycle [35].

once again. The complete cycle is illustrated in Fig.3.3. The operating economics of the desiccants depends on the energy of moving a given material through this cycle. The dehumidification of air generally proceeds without energy input, other than fan and pump costs. The major part of the energy is invested in regeneration of the desiccant (state 2 to state 3) and cooling the desiccant (state 3 to state 1). Regeneration of energy is equal to the sum of three variables [35]. The heat necessary to raise the desiccant to a temperature high enough to make its surface vapor pressure higher than the surrounding air, the heat necessary to vaporize the moisture it contains, and the amount heat for desorption of water from the desiccant.

The ideal desiccant for a particular application depends on the range of water vapor pressures that are likely to occur in the air, the temperature level of the regeneration heat source, and the moisture sorption and desorption characteristics of the desiccant within those constraints. The useful life of desiccant materials depends largely on the quantity and type of contamination in the airstreams they dry. In commercial equipment, desiccants last between 10,000 and 100,000 hours and longer before they need replacement [35]. Normally, two mechanisms cause the loss of desiccant capacity, change in desiccant sorption characteristics through reactions with contaminants, and loss of effective surface area through clogging or hydrothermal degradation. Liquid absorbents are more susceptible to chemical reaction with airstream contaminants other than water vapor than unlike solid adsorbents. In air-conditioning applications, desiccants equipment is designed to minimize the need

for desiccant replacement in much the same way that vapor compression cooling systems are designed to avoid the need for compressor replacement. Unlike filters, desiccants are seldom intended to be frequently replaced during normal service in an air-drying application.

3.5 Desiccant Cooling System Components

The major components in a desiccant cooling systems are: dehumidifier, regenerative heat exchanger, direct or indirect evaporative coolers, fans, heaters, and control hardware. In addition to these components, advanced desiccant cycles use cooling towers and hybrid desiccant systems use vapor compression components. Most of the components, except the desiccant dehumidifier, have been evolving for many years within other technologies. For example, relatively inexpensive and efficient heat exchangers, evaporative coolers, and fans are available commercially. Therefore, the desiccant dehumidifier has received the most attention over last 15 years of desiccant cooling research and development. Recent research and development efforts on dehumidifiers have focussed on improving their performance and reliability and reducing their cost and size.

The desirable characteristics for high-performance liquid desiccant dehumidifiers are

1. High heat and mass transfer rates.

2. Low pressure drops.
3. Small liquid side resistance to moisture diffusion.
4. Large contact transfer surface area per unit volume.
5. Compatible desiccant / contact materials.
6. Use of common materials and inexpensive manufacturing techniques.
7. Low heat input for regeneration.

By combining a desiccant dehumidifier to meet the latent load and vapor compression or vapor absorption system to meet the sensible load, a very efficient system can be built. The dehumidifier in the hybrid system is required to remove less moisture in the desiccant system. This allows the desiccant to be regenerated at lower temperatures; e.g., by using solar energy more efficiently [37]. Also waste heat can be used; e.g, from condenser of the vapor compression unit, from a thermally activated heat pump, or from cogeneration systems. The vapor compression unit has to handle only the sensible load, so a smaller unit may be used. Performance of the unit is also improved by at a higher evaporator temperature because it does not have to condense moisture. In supermarkets or food store, much of the sensible cooling is provided by freezer and refrigeration cases; however, additional dehumidification is always required. By using a desiccant dehumidifier to remove this moisture, the size of the vapor compression unit and the electrical power consumed can be sub-

stantially reduced. Additionally, the freezer and refrigeration cases can be operated more efficiently because of reduced frost buildup.

3.6 Hybrid Systems

A hybrid desiccant system uses a desiccant (solid or liquid) to reduce the humidity ratio to bring the temperature within the comfort range. In most of the hybrid systems, the heat rejected from the condenser of the vapor compression (V.C) cycle or vapor absorption (V.A) cycle is used partially to regenerate the desiccant. This results in the improvement of the overall COP of the system. The moisture content of air is reduced by the desiccant cycle. This will partially convert the latent heat cooling load into sensible heat cooling load, and then meeting this load with evaporators of V.C or V.A systems. Therefore, in hybrid systems, the operating temperature in a V.C or V.A system evaporator is much higher, thereby further improving the overall COP of the system for space conditioning [24]. Hybrid systems offer the advantages of reduced energy costs and reduced equipment size and independent control of humidity and temperature [7]. The overall COP of the hybrid desiccant systems predicted by the investigators ranges from 0.5 to 1.6. The hybrid system, with solid desiccant for latent heat load and vapor compression system for sensible cooling, resulted in 25% energy savings over a standard vapor compression cycle. Similarly, a 35% energy saving was observed by coupling a liquid desiccant dehu-

midifier to a vapor compression cycle [24]. Calculations have shown a better savings in coupling a desiccant (solid/liquid) dehumidifier to a vapor-absorption cycle.

3.6.1 Hybrid Solid Desiccant Systems

In this system a $LiBr - H_2O$ or a $H_2O - NH_3$ absorption machine is coupled to a solid desiccant (silica gel) dehumidifier. The heat released in the absorption system condenser is used for partially regenerating the desiccant. Remaining part of the regeneration process is carried out with the use of any low grade thermal energy. The absorption system meets the sensible load, while the solid desiccant system takes care of the latent load. Since the absorption machine has to meet only the sensible load, it is smaller than the conventional absorption machine which has to meet both the sensible and latent loads simultaneously. The refrigeration system consists of a hot, concentrated solution in equilibrium with the condenser pressure leaving the generator and being throttled and cooled to the absorber pressure. The cold, concentrated solution absorbs low pressure refrigerant in the absorber in equilibrium with evaporator pressure. The weak solution enters the generator, where heat is added to distill the refrigerant. The hot, high-pressure refrigerant vapor condenses in the condenser. Hot, liquid refrigerant is expanded into the evaporator, where it is evaporated at low-pressure and temperature with heat from the cold space. The cold, low-pressure refrigerant vapor is absorbed by the solution in the absorber and the cycle is completed.

The dehumidifier system consists of air passing through the desiccant (silica gel) in the dehumidifier, where it is dehumidified. The desiccant is regenerated with the heat released in the absorption system condenser. The air then passes through the evaporator coil of the absorption machine where its sensible load is met. A combined overall COP of 1.55 was observed from the above system which is about 77%, 83%, and 89% more than the expected COP from the desiccant, Rankine and absorption solar cooling systems [4]. A condenser temperature of 75⁰ C was used to regenerate the silica gel dehumidifier.

A system similar to the above mentioned system with a vapor compression cycle for the sensible cooling load is used with the solid desiccant for latent cooling load [4]. In this system the heat rejected from the condenser of the vapor compression cycle is used partially to regenerate the desiccant, thereby improving the COP of the system. Further regeneration of the desiccant is provided with auxiliary heat sources/flat plate solar collectors etc. The regeneration temperature of these auxiliary sources should be just sufficient to meet the dehumidification required.

The combination of a V.C system with a desiccant unit resulted in a 60% reduction in air conditioning costs as compared to a single standard V.C system for the same loads [24]. It is found that a further reduction in air-conditioning costs can be observed by using proper heat exchangers with high effectiveness within the system. But these systems have the disadvantage of using CFCs. The desiccant performance is appraised by defining the effectiveness parameters, $effdx1$ and $effdx2$ for a par-

ticular wheel. For $effdx1 = 0.07$ and $effdx2 = 0.4$ and a regeneration temperature of 85°C , the resulted thermal desiccant COP was 1.15. The COP reduces to 0.74 due to air leakage of 2% at the wheel circumference and 4% across the air stream with an increase in regeneration temperature of 90°C [27].

3.6.2 Hybrid Liquid Desiccant Systems

Liquid desiccant systems have not been extensively investigated in the past because of the COP values predicted were below 1.0. However, liquid desiccants have some advantages over solid desiccant systems. Because the energy is stored in the form of chemical energy rather than thermal energy, the reliance on continuous thermal energy supplies is reduced, and the amount of energy stored is greater. Liquid desiccants also offer greater design flexibility than solid desiccant systems because the components can be installed in different locations and the liquid pumped between them. The disadvantage of liquid desiccant systems can be their size at low capacities. Liquid desiccants can have either carryover problems or corrosion problems if not designed properly [37].

In this system, a $\text{LiBr} - \text{H}_2\text{O}$ or $\text{H}_2\text{O} - \text{NH}_3$ absorption machine is coupled to a liquid desiccant dehumidifier. The advantages of using a liquid desiccant is that it does not require complex dehumidifier geometries, as the desiccant is usually sprayed over the incoming air and can be regenerated on inexpensive open-flow collectors or from the heat of the absorption machine condenser.

This system uses a cooling tower to cool both the absorption machine absorber and the desiccant dehumidifier. Since the absorption chiller has to meet only the sensible load, it is smaller than a conventional chiller. The mathematical modelling of the system has resulted in COP values of around 0.9 to 1.2 depending on the indoor humidity ratio [7]. The cycle is similar to that of a hybrid solid desiccant system with the liquid desiccant being circulated between the condenser of the absorption machine and the auxiliary heater or flat plate solar collector. A similar system with a vapor compression cycle for sensible load and a liquid desiccant unit for latent load can be used [24]. This system consists of a liquid desiccant being circulated between the condenser and the evaporator of a vapor compression cycle. The condenser is used for partial desiccant regeneration and the evaporator surface is used as an absorber for liquid desiccant cycle. Lithium bromide, chloride or Calcium chloride is used as the liquid desiccant. In this system, the efficiency of the dehumidification will be very high, since the desiccant is cooled to the lowest available system temperature, that is evaporator temperature, thus decreasing the desiccant vapor pressure to its lowest possible value.

Many simulation techniques have been devised to estimate the performance of these hybrid systems. A computer programme BINSYS, which takes into account the BIN system can be used to evaluate the performance and to aggregate the results of an entire year [27]. The performance of a hybrid system is related to the difference of enthalpy of room and the supply air. The cooling effect delivered

depends on both the heat exchangers' effectiveness and on the degree of dryness of the air leaving the dehumidifier, which in turn depends on the outdoor conditions and on the regeneration temperature.

Chapter 4

HYBRID CYCLE ANALYSIS

4.1 Introduction

Comfort air-conditioning involves treating the environmental air to control its temperature, humidity, cleanliness, and distribution for the comfort of the occupants of the air-conditioned space. In hot climate areas, an aggressive environment, such as a salt-laden atmosphere or frequent sand storms can cause a sudden change in the environmental conditions. Hence the air has to be both sensibly and latently cooled, with primary consideration being given to salt and dust free air that has to be supplied to the comfort space. In these hot climate areas solar energy has been proved to be a source which is available in abundance and free of operating cost. Conventional air-conditioning systems adjust both temperature and humidity ratio to the desired values by passing outdoor air through cooling coils. The air is

cooled below its dew point so that water vapor condenses to meet the indoor air specifications. For a typical humidity ratio of 0.007, this occurs at 9°C, which is usually below the desired temperature. Therefore, the air has to be reheated. Although this is generally done by using free waste heat or by mixing with return air, the cooling process itself requires more energy than a thermodynamically optimal process with a direct path from the outdoor air state to indoor air state. Hybrid desiccant cooling systems can follow this direct path more closely by splitting up the air-conditioning task into cooling (sensible load) and dehumidification (latent load). Absorption and desiccant systems appear to be the most promising solar-powered cooling arrangements. In both systems solar energy is employed for the process of regeneration. In the former system, the solar energy is used to heat the absorption machine generator (solar collector). In a desiccant system, however, a liquid desiccant removes the moisture from the air in a dehumidification chamber. Thereafter, the moisture-loaded desiccant is regenerated using solar energy to drive off the moisture from the desiccant material to repeat the dehumidification cycle.

4.2 Comparison of Absorption and Vapor Compression Systems

The vapor compression and vapor absorption refrigeration machines are shown in Figures 4.1 and 4.2. It is unfair to the absorption system to make a direct comparison

of its COP with that of a mechanical compression system. A COP calculated for an absorption system is more realistic than a COP calculated for a theoretical vapor compression cycle where isentropic compression is assumed. In order to make a comparison, we must multiply the actual COP of the mechanical compression system by its thermal efficiency. We must also include the transmission losses up to the compressor. When we do this, we find that a mechanical compression system has little thermodynamic superiority over an absorption system.

Apart from COP considerations the absorption system has some practical advantages over mechanical compression systems. The absorption system is less subject to wear and maintenance. The absorption system may operate at reduced evaporating pressure with little decrease in refrigerating capacity. Liquid carry over from the evaporator causes no difficulties. The choice between an absorption system and a mechanical compression system is largely determined by the economic factors. The absorption water chilling machine used for air conditioning may be economically attractive in areas where low cost fuel is available, electricity rates are high, heating boiler capacity is idle or partially idle during summer months, waste steam is available, and existing electric facilities are inadequate for installing electric motor driven mechanical compressors [38].

There are similarities and differences between the vapor compression and absorption systems. The useful refrigeration is achieved in the same way in both systems-by the evaporation of a liquid in an evaporator, using the latent heat of vaporization

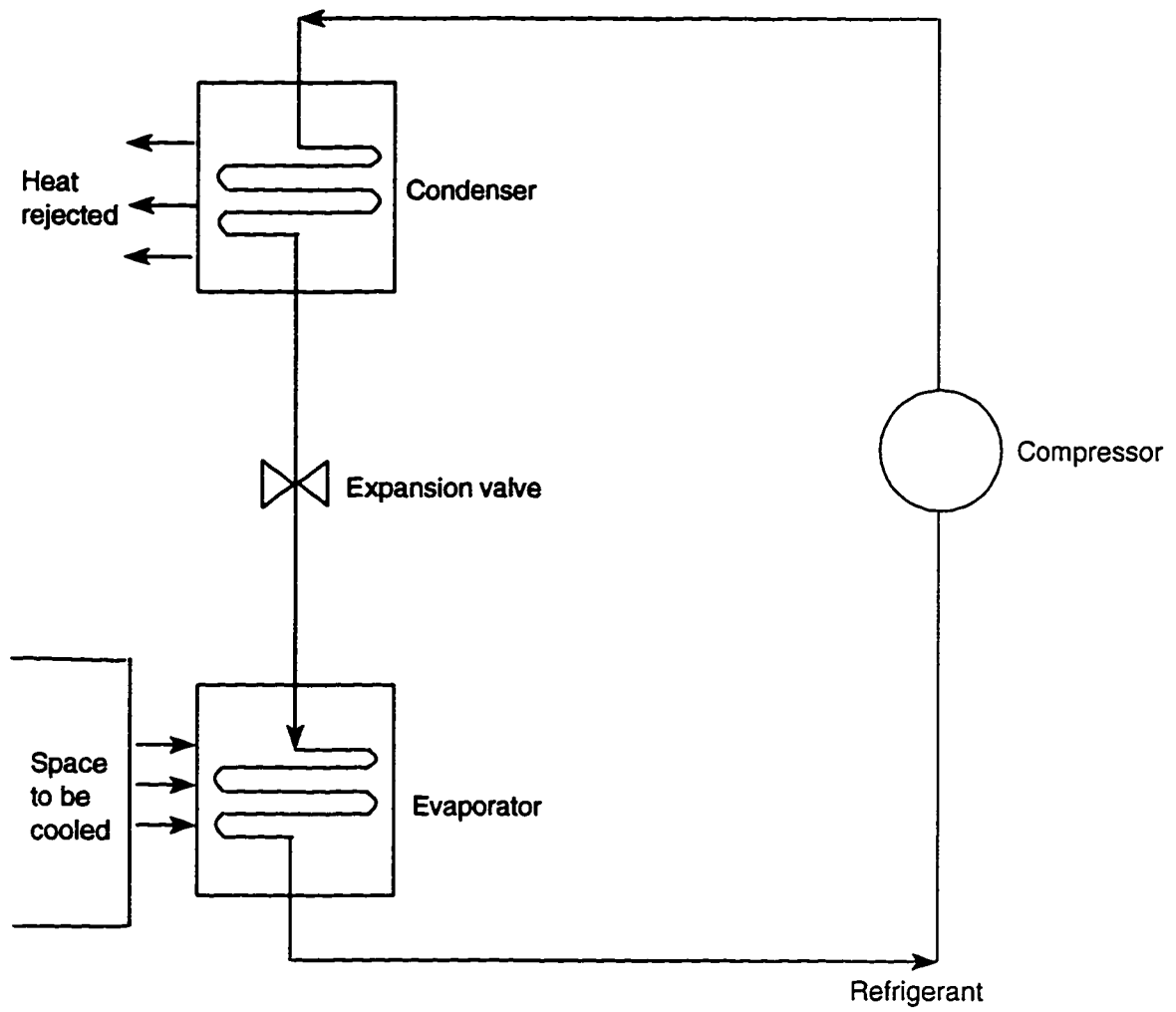


Figure 4.1: Vapor Compression Refrigeration Cycle.

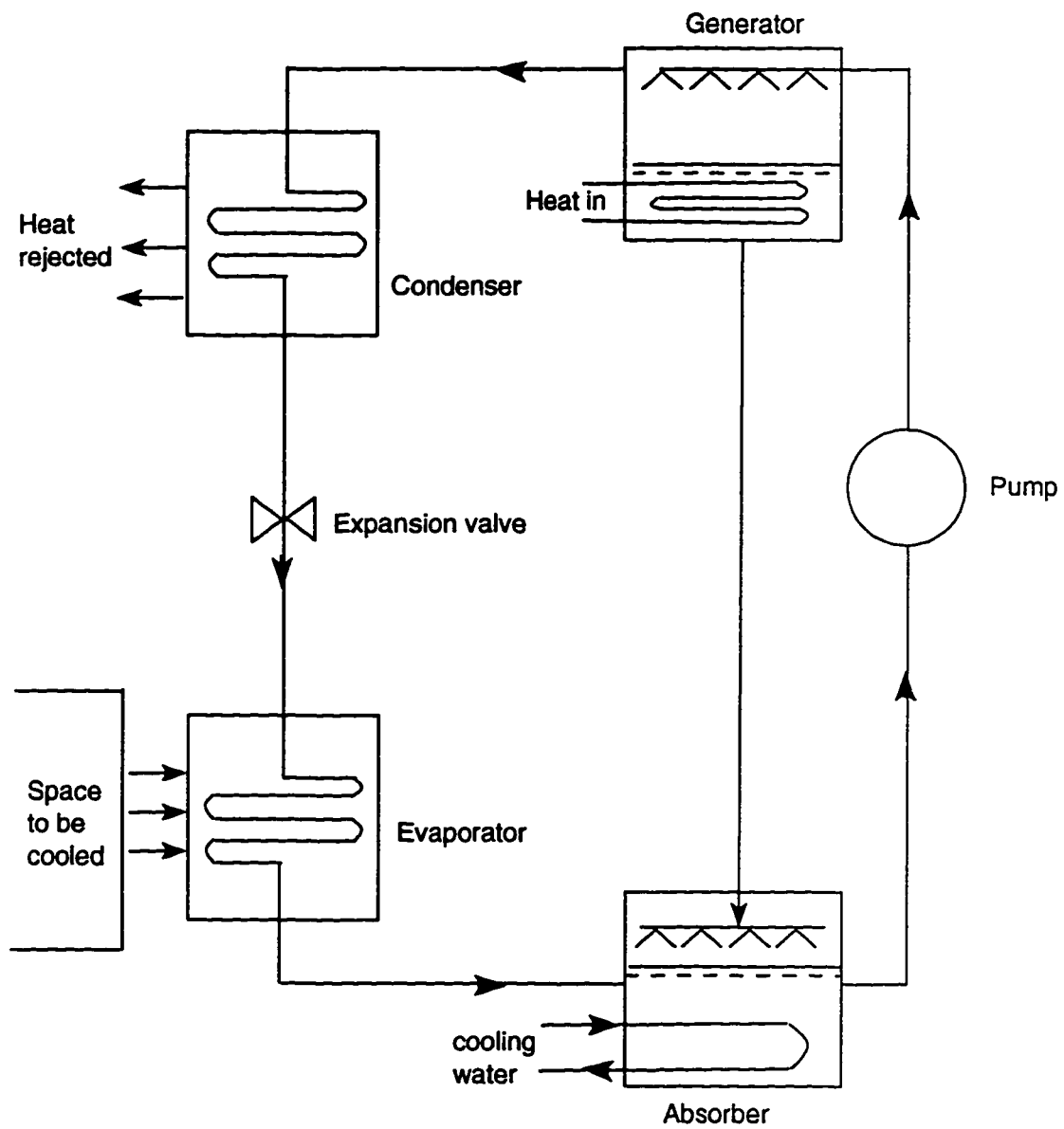


Figure 4.2: Vapor Absorption Refrigeration Cycle.

to achieve the cooling effect. Both systems can also use a condenser, to remove heat from the high pressure refrigerant vapor, and thus return it to its original liquid state. They both use flow control or expansion device. The system differs, however, in the way by which the evaporated refrigerant is recovered and how its pressure is increased [39]. In the first place, the forms of energy used to operate the system are different. In vapor compression system, mechanical energy is used to drive a compressor. The operation of the compressor maintains the low evaporator pressure and also raises the pressure to high-side pressure. The comparison of the COPs of the two conventional refrigeration systems will give an idea as to how much the performance of these two systems can be improved after combining them with the desiccant cycle.

4.3 Absorption Air-Conditioning System

In the absorption system, heat energy is used to raise the refrigerant pressure. Low evaporator pressure is maintained by the use of a second substance called an absorbent. Two components, the absorber and the generator, serve a function similar to the compressor. Often auxiliary components such as pumps are used in the absorption systems. One reason the absorption system is popular and versatile is that it operates directly on heat energy [40]. Wherever waste steam, hot water, or combustion gases are available, absorption refrigeration is a good candidate for

consideration there.

4.3.1 The Absorption Process

Some pairs of substances have an affinity or attraction for each other such that when they come in contact one becomes absorbed by the other. The substance that absorbs the other is called the absorbent. Absorbents may be solids, liquids, or gases, and substances for which they have an affinity may be liquids or gases. In absorption refrigeration, the absorbent is a liquid and the other substance, which serves as the system refrigerant is in a gaseous (vapor) state when it is absorbed. Two pairs of substances have been successfully used in absorption systems. One pair is water and ammonia. Water is the absorbent and ammonia is the refrigerant. The other pair is lithium bromide and water. In this case lithium bromide is the absorbent and water is the refrigerant.

The mixture of the two substances is normally in a liquid state in refrigeration systems. When the proportion of absorbent in the mixture is high and the proportion of the refrigerant is low, it is called a concentrated or strong solution and when the proportion of absorbent is low and the proportion of the refrigerant is high, it is called a dilute or weak solution. Concentration is usually expressed as the percentage by weight of the absorbent. A more concentrated solution can absorb a greater quantity of refrigerant, and will do it at a faster rate. There is maximum amount of refrigerant that a solution can absorb, when it becomes too dilute it will

no longer absorb any refrigerant.

4.3.2 The Absorber and Evaporator

The evaporating temperature of the refrigerant depends on its saturation pressure temperature characteristics. That is, the temperature at which the refrigerant will vaporize will vary with the evaporator pressure. The pressure must be low enough so that the resulting evaporating temperature will be a value needed for the refrigeration application. The evaporator contains the refrigerant (water). The absorber is an adjoining vessel containing a strong solution (lithium bromide-water). The vessels are connected so that vapor may flow freely between them. The vessels are evacuated of an air to an extremely low pressure. Water in the evaporator will begin to flash to a vapor, that is, it will boil. The cooling effect of the latent heat of vaporization will cool the unevaporated water, achieving refrigeration. The water vapor produced will fill the evacuated space in both vessels. The pressure in the evaporator would continually increase as more refrigerant evaporates. The corresponding evaporating temperature would therefore also increase, and refrigeration at an acceptably low temperature would be lost. This is prevented by the action of the absorbent, however. The water vapor filling the vessels will contact the surface of the strong absorbing solution in the absorber, and will be absorbed by it. This reduces the pressure in the space. A slight pressure gradient is created from the surface of the evaporating refrigerant to the absorber, which promotes the flow of

water vapor. That is the absorber will draw off the water vapor as it is produced, thereby maintaining the low pressure and temperature in the evaporator. This action of the absorber replaces the suction effect created by the compressor as in vapor compression systems.

Since the evaporator has to be kept at very low pressure the equipment must be sealed very tightly. This would be physically impractical if the refrigerant water was circulated externally through piping and coils to the load. The refrigerant is used to chill the water circulating in a tube bundle in the evaporator. Regardless of absorption machine's physical arrangement, one major factor must be considered, which is, that the vessel or vessels must be held at high vacuum. Any air leakage into the system will reduce the machine capacity, since the evaporation of the water relies of the high vacuum to produce low boiling temperatures. Secondly, there is an increased possibility that solution crystallization will occur. The reduction in capacity reflects lighter loads on the generator such that the solution will be concentrating at high temperatures and low pressures with resulting over strength solutions. If the solution is highly concentrated it may crystallize when cooled in the solution heat exchanger. The third consideration of air leakage is one of corrosion. A vacuum-tight system is oxygen free, and oxygen with the lithium bromide solution produces a corrosive action which will attack the metal components of the system. The maximum temperature of 90/10 copper nickel surfaces contacting lithium bromide solutions (to avoid excessive corrosion) is about 160 ° C [41].

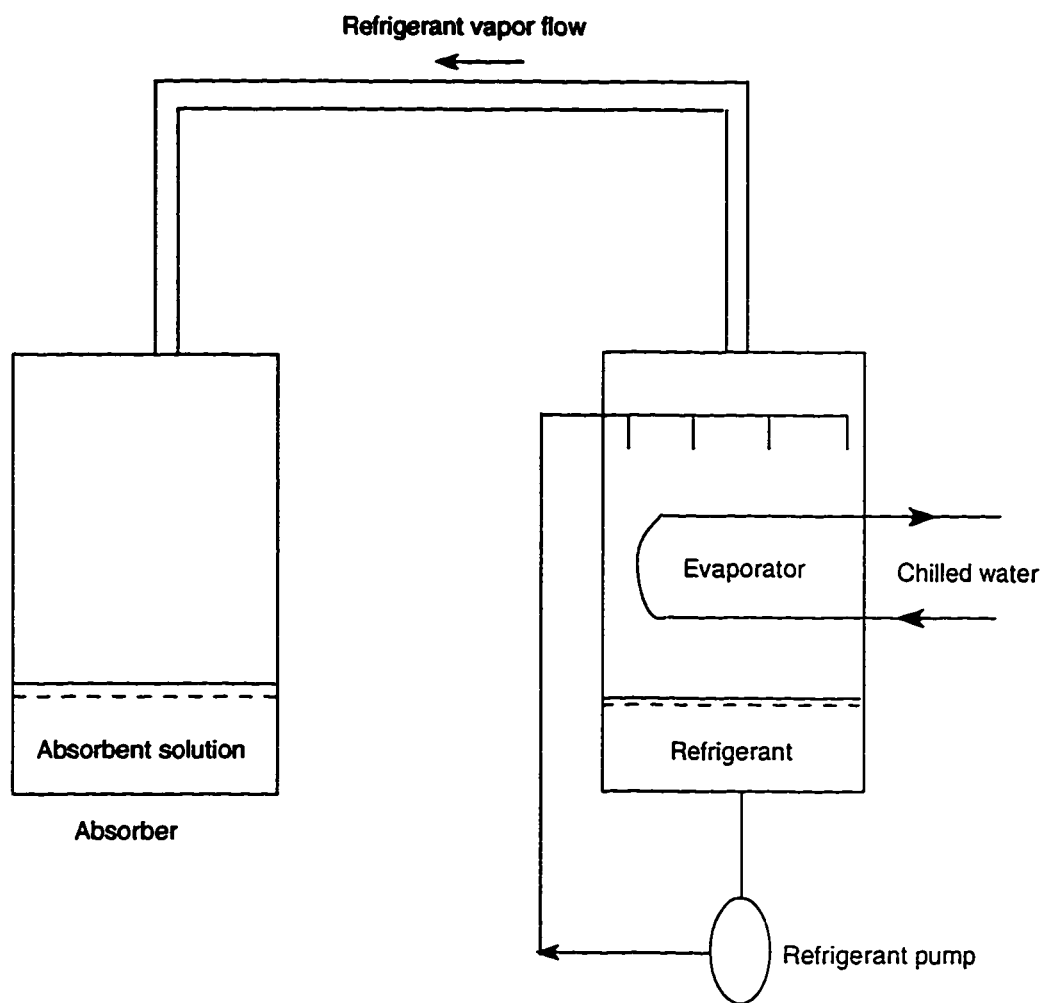


Figure 4.3: The Refrigerant pump and the Chilled Water Distribution Coil Added to the Evaporator.

All lithium bromide absorption systems must have an efficient purge system to overcome the problem of air leakage. As with the absorption unit arrangement the method of purging has been developed from the early machines which had two stage steam and water jet systems with development to a self contained purge system, which had its own water pump; this was developed further to another self contained system using a single jet of lithium bromide as purging medium. The single stage purge was possible because of the lower solution vapor pressure which increased the purging capacity of standard ejectors. The chilled water is then circulated to the load. The refrigerant water can also be continuously recirculated by an evaporator pump through spray header nozzle into the chilled water tubing as shown in Fig. 4.3. A pump and spray nozzle arrangement is also used to spray the strong absorbent solution into the absorber space. This increases the surface area contact between the solution and the water vapor, increasing the effectiveness of the absorption process [38]. The process of absorption in the absorber would not operate satisfactorily for very long because the absorbing solution will become more dilute as it absorbs water vapor and soon would be ineffective. The rate of absorption of water vapor would decrease, raising the pressure and evaporating temperature to an unacceptable level. At some point the absorbent solution would stop absorbing water vapor completely.

4.3.3 Recovery of the Refrigerant

Obviously some means of recovery of the refrigerant must be arranged for, if the system is to be used more than once. Two basic steps are needed. First, the refrigerant must be separated from the solution. This is accomplished by heating the weak solution to a temperature at which the water boils off from it in a component called the generator. Second the water vapor must be condensed to a liquid (closed cycle) so that it is ready for use in the evaporator again or lost to the atmosphere (open cycle). The weak solution is pumped from the absorber to the generator. Heat is applied to the solution, resulting in the boiling of the water vapor [42]. Now, the reconcentrated strong solution is returned to the absorber and sprayed into the surrounding refrigerant water vapor, where the absorption takes place. The heat source is often steam, hot water or solar energy.

When the lithium bromide solution absorbs water, heat is generated which consists of heat of condensation of water plus a reaction heat between the lithium bromide and water vapor. To increase the capacity of lithium bromide to accept the water vapor it should be kept cool; this is achieved by passing the cooling water through the absorber. Because the generator is hot and the absorber cool, the cycle efficiency can be increased by incorporating a heat exchanger which heats the weak solution pumped from the absorber to the generator and cools the strong solution returning from the generator.

4.3.4 The Heat Exchanger and Cooling Water Circuits

An energy saving feature that is included in the actual system is a heat exchanger provided between the weak and strong solutions. The weak solution from the absorber is preheated by hot solution leaving the generator. This saves part of the energy to bring the weak solution up to boiling point temperature, which would otherwise come from the heat source. The cooling of strong solution in the heat exchanger also reduces the cooling requirement in the absorber to absorb the refrigerant at proper operating temperatures. The heat that is generated in the absorber must be removed.

There are three sources of heat released in the absorber. First, the refrigerant (moisture) vapor when it is absorbed becomes liquid. The latent heat of condensation of this effect must be removed. Second, the absorption process itself generates heat from chemical effects, called heat of dilution. Finally, despite the heat exchanger, further sensible heat must be removed from the returning strong solution to bring it down to operating temperature [42]. The cooling water is circulated to the absorber. Any of the usual sources of cooling water, such as a cooling tower, may be used.

4.3.5 Absorbent and Refrigerant Characteristics: Crystallization

Both the absorbent and refrigerant should have certain desirable characteristics. The lithium bromide-water pair are quite satisfactory in most respects. They are low cost, chemically stable, and nontoxic when compared to the ammonia-water pair. Lithium bromide will absorb large quantities of water. Therefore only a relatively small amount of lithium bromide must be pumped between the absorber and generator. The vapor pressure-temperature characteristics of water are such that pressures in the vessels must be extremely low. This is undesirable, since air will leak in through any poorly sealed joints [38]. This would raise the pressure and the resulting increased evaporating pressure which would be unacceptable. The application of lithium bromide -water machines are limited to relatively high temperatures, since the refrigerant freezes at 0°C .

The lithium bromide-water solution has one property that can cause difficulty. If a strong solution at a high temperature is cooled solid crystals will precipitate out of liquid. This process is called crystallization. Crystallization of salts means that the crystals formed are 100 percent stale, which weakens the balance of the solution which has not crystallized, and therefore it is until the solution becomes 100 percent salt that complete solidification occurs. Crystallization will cause no damage to the heat exchanger parts since, unlike water, it contracts on solidification; however

crystallization will make solution pump inoperative and satisfactory design demands that cycle operation be maintained outside the crystallization zone. The resultant mixture has consistency like slush. If crystallization occurs in an absorption machine, refrigeration will cease, since the pumps cannot move the slushlike mixture around.

4.4 The Hybrid Cycle

The proposed hybrid cycle consists of a liquid-desiccant dehumidifier integrated with an absorption cooling system and uses *LiBr* as the liquid desiccant for both absorption and dehumidification processes and water as refrigerant. The system operates on an open-cycle in which the weak absorbent solution is regenerated by losing the refrigerant to the earth's atmosphere [13, 14]. Therefore, the condenser is eliminated in the open-cycle system unlike the closed-cycle system in which the refrigerant is condensed in the condenser before throttling it to the evaporator conditions. Cooling takes place by evaporating refrigerant from an external source in the evaporator rather than obtaining refrigerant from the condenser. However an external source of refrigerant (water) must always be available [15].

The weak absorbent solution (state 1) from the absorber is pumped (state 2) to the solar regenerator which is partly closed-open to the atmosphere through the regenerative heat exchanger HE1. It is then heated and subsequently concentrated in the solar regenerator from state 3 to state 4. The strong solution passes through

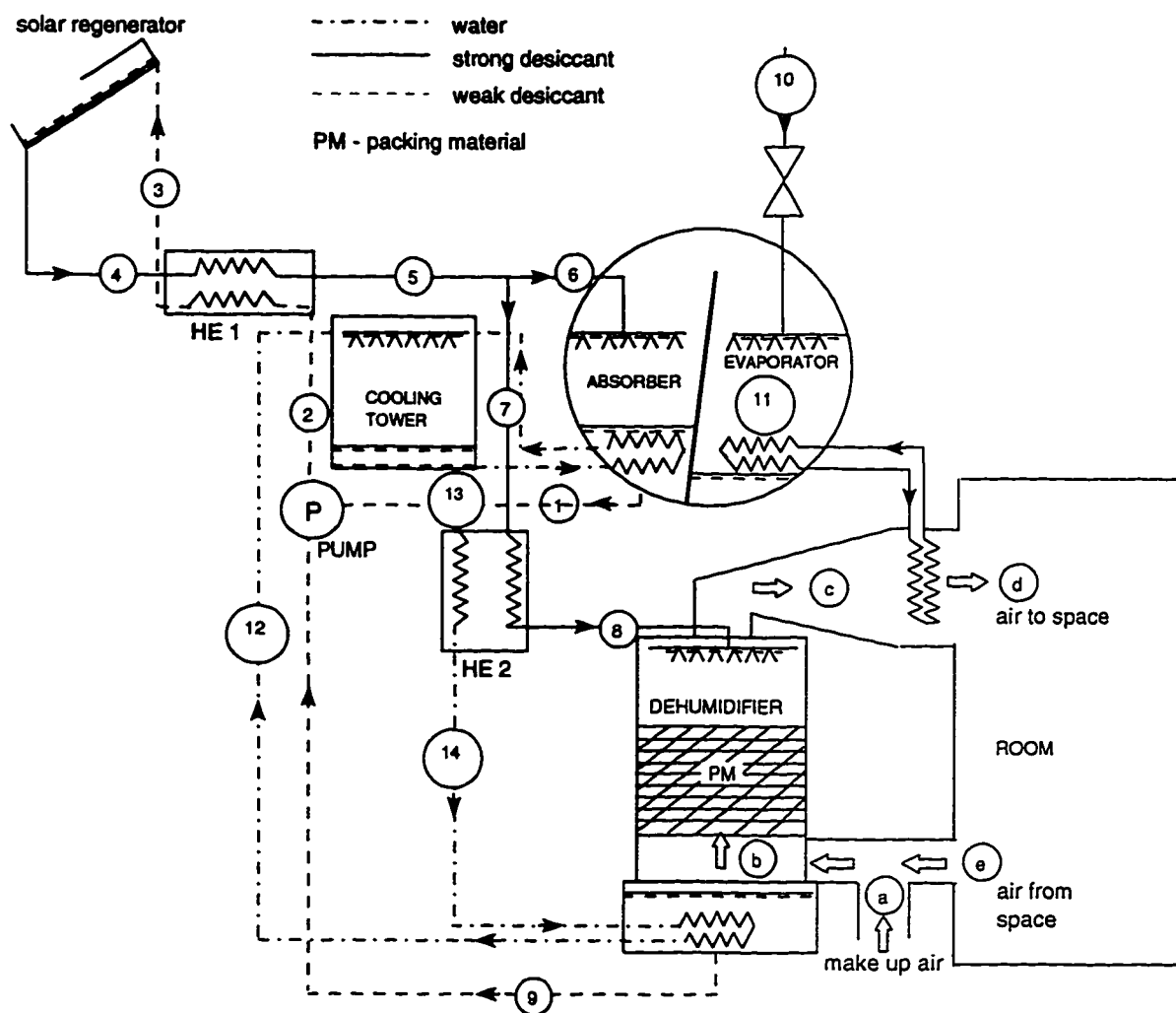


Figure 4.4: Hybrid Liquid Desiccant Based Air-Conditioning Cycle.

the regenerative heat exchanger HE1 on its way (state 5 and 6) to the absorber. In the absorber, the strong desiccant absorbs water vapor as soon as the incoming refrigerant evaporates in the evaporator (state 11) by absorbing the heat from the cold space, maintaining the reduced pressure required by the evaporator. The heat of absorption for the refrigerant-absorbent solution is removed by cooling water loop. In evaporator, water from an external source (state 10) is evaporated at reduced pressure with the energy supplied by the heat from the cooled space (states 15 and 16). The resultant weak desiccant is then pumped from the absorber back to the atmospheric pressure through the regenerative heat exchanger and to the solar regenerator, completing the absorption-machine cycle.

Part of the strong desiccant (state 5) from the solar regenerator is passed through the regenerative heat exchanger HE2 (state 7) to the dehumidifier, where the air to be conditioned is dehumidified in the counter-current direction. The actual dehumidification takes place due to the vapor pressure difference between the vapor in the air and the liquid desiccant. As the regenerated liquid desiccant is cooled and concentrated (state 8), its vapor pressure is less than the vapor pressure of the air, therefore the vapor present in the air tends to escape into the liquid desiccant, thereby diluting the liquid desiccant. The weak desiccant (state 9) from the dehumidifier is then pumped back for regeneration along with the weak desiccant from the absorber (state 1). After regeneration the strong solution is supplied to absorber and the dehumidifier in the required proportions as shown by states 6 and 7. The

absorbent used in the hybrid cycle for absorption in absorption machine as well as the desiccant in the dehumidifier is lithium bromide. The cooling water (states 12, 13, 14, 17 and 18) is circulated to the dehumidifier and completes the cooling tower cycle.

4.4.1 Mass and Energy Balance

Referring to Fig.4.4, a total mass balance on the regenerator gives,

$$\dot{m}_3 = \dot{m}_t + \dot{m}_4 \quad (4.1)$$

where \dot{m}_t is the total moisture (refrigerant and dehumidified moisture) and is given by,

$$\dot{m}_t = \dot{m}_r + \dot{m}_{rd}$$

Whereas LiBr mass balance yields,

$$\dot{m}_3 X_3 = \dot{m}_4 X_4 \quad (4.2)$$

Solving Eq.(4.1) and Eq.(4.2), we get

$$\frac{\dot{m}_4}{\dot{m}_t} = \frac{X_3}{(X_4 - X_3)}$$

This gives the ratio of total absorbent-to-total refrigerant flow rate. It is assumed that exactly 50% of the concentrated solution from the regenerator is supplied to the absorber and the remaining 50% to the dehumidifier. Therefore, since $\dot{m}_4 = 2\dot{m}_{ab}$, the ratio of the total refrigerant-absorbent solution (solution from absorber and

dehumidifier) flow rate to the refrigerant (refrigerant and moisture from the dehumidifier) solution flow rate is given by,

$$\frac{\dot{m}_{st}}{\dot{m}_t} = \frac{2\dot{m}_{ab}}{\dot{m}_t} + 1$$

where \dot{m}_{st} is the total mass of the solution reaching the solar regenerator and is given by

$$\dot{m}_{st} = \dot{m}_s + \dot{m}_9$$

\dot{m}_s is given by the expression,

$$\frac{\dot{m}_s}{\dot{m}_r} = \frac{\dot{m}_{ab}}{\dot{m}_r} + 1$$

The heat balance for the heat exchangers HE1 and HE2 gives

$$\dot{m}_2 h_2 + \dot{m}_4 h_4 = \dot{m}_3 h_3 + \dot{m}_5 h_5, \text{ and} \quad (4.3)$$

$$\dot{m}_7 h_7 + \dot{m}_{13} h_{13} = \dot{m}_8 h_8 + \dot{m}_{14} h_{14} \quad (4.4)$$

The evaporator capacity is given by

$$Q_{ev} = \dot{m}_{11} (h_{11} - h_{10}) \quad (4.5)$$

The mass balance in dehumidifier results in

$$\dot{m}_{rd} + \dot{m}_8 = \dot{m}_9 \quad (4.6)$$

With 40°C and 2 kPa as the equilibrium conditions of the absorber, the weak LiBr solution leaves the absorber with a concentration of 50 percent and an enthalpy of 80

kJ/kg [4]. Using a partly closed-open solar regenerator with an outlet temperature of about 80°C, the equilibrium concentration at the regenerator conditions will be 55 percent and the strong solution leaves the regenerator with an enthalpy of 180 kJ/kg. Hence the ratio of the absorbent \dot{m}_4 -to- refrigerant \dot{m}_t flow rate and the ratio of the refrigerant-absorbent flow rate \dot{m}_{st} to the refrigerant flow rate \dot{m}_t are 10 and 11 respectively, [43], [44].

Individual component of the hybrid cycle is shown in Figs. 4.5, 4.6, 4.7, 4.8, 4.9. Energy balance for each individual part for sizing the different components of the system is performed. The amount of solar energy collected for the regeneration of the weak desiccant is given by,

$$Q_{gen} = \dot{m}_t h_t + \dot{m}_4 h_4 - \dot{m}_{st} h_3 \quad (4.7)$$

The absorber load is given by

$$Q_{ab} = \dot{m}_{11} h_{11} + \dot{m}_6 h_6 - \dot{m}_1 h_1 \quad (4.8)$$

The dehumidifier load is given by

$$Q_{rd} = \dot{m}_{rd,i} h_{rd,i} + \dot{m}_8 h_8 - \dot{m}_9 h_9 - \dot{m}_{rd,o} h_{rd,o} \quad (4.9)$$

The total regenerator loss is given by

$$Q_{gl} = \dot{m}_t (h_t - h_3) \quad (4.10)$$

The hybrid cycle parameters are calculated at every state points of the cycle by using the empirical expressions shown in Appendix I and Appendix II. Referring to

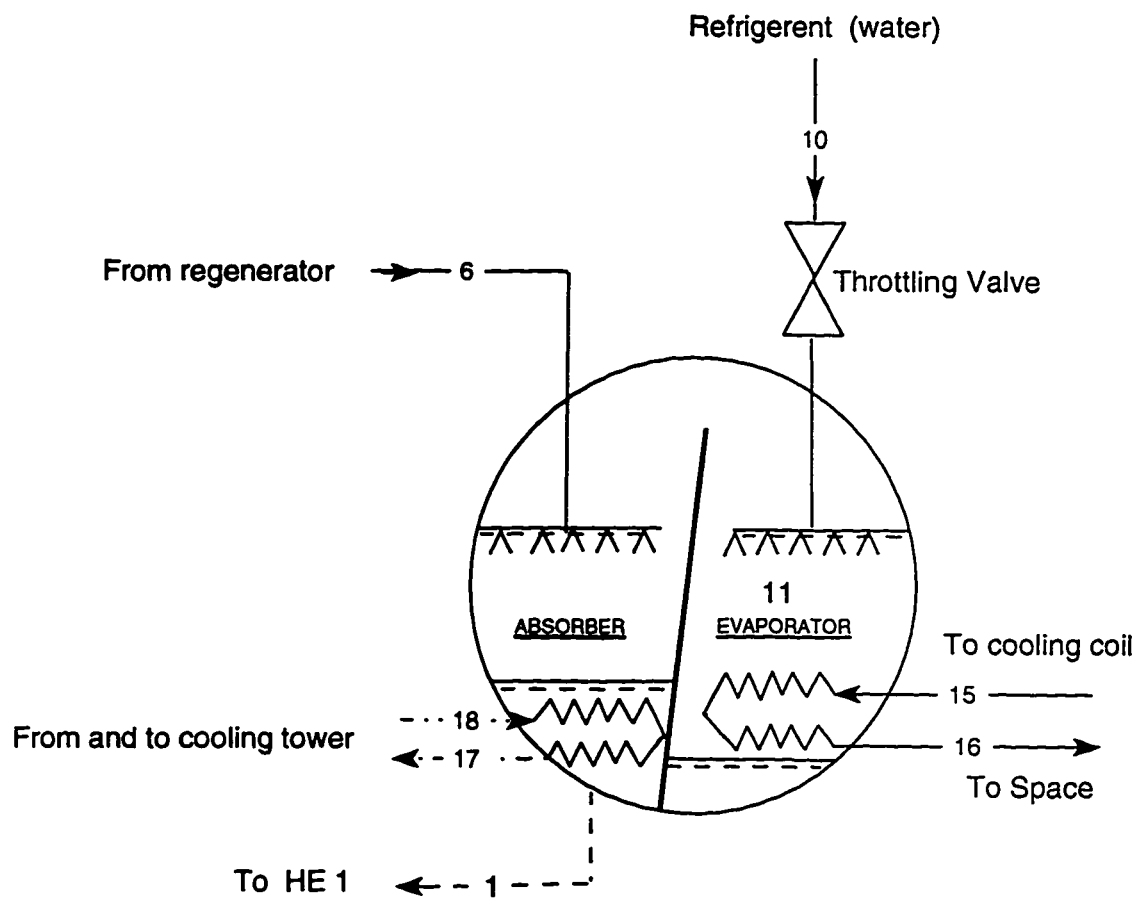


Figure 4.5: Hybrid Cycle Components: Absorber and Evaporator.

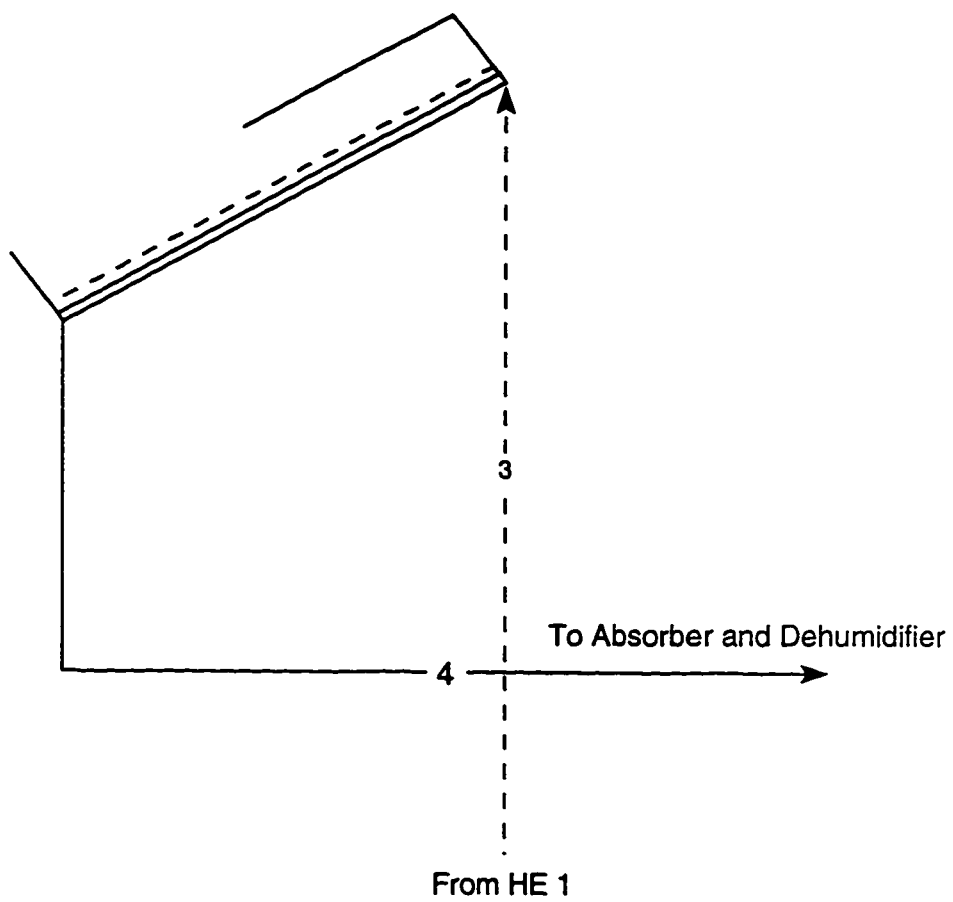


Figure 4.6: Hybrid Cycle Component: Regenerator

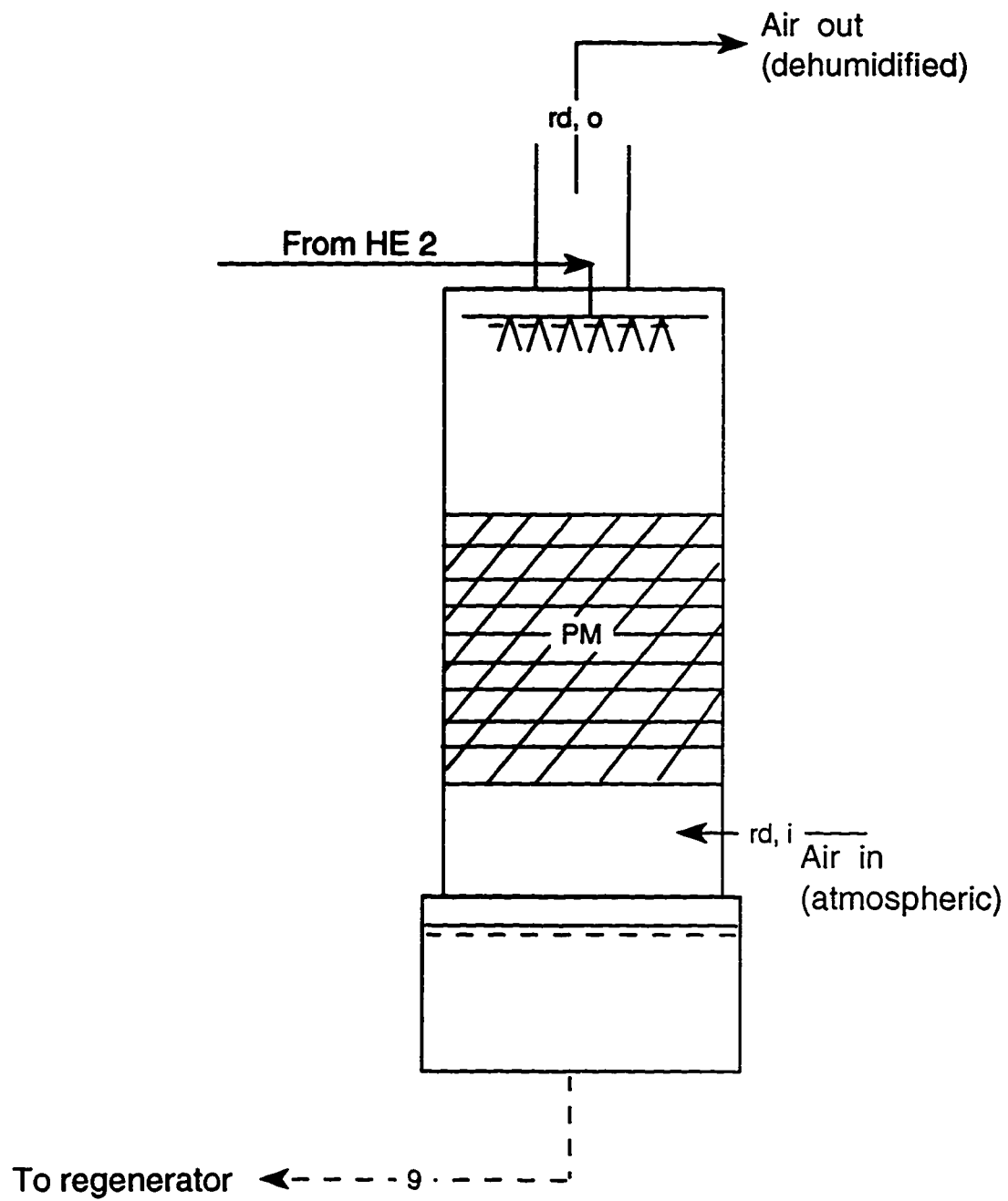


Figure 4.7: Hybrid Cycle Component: Dehumidifier

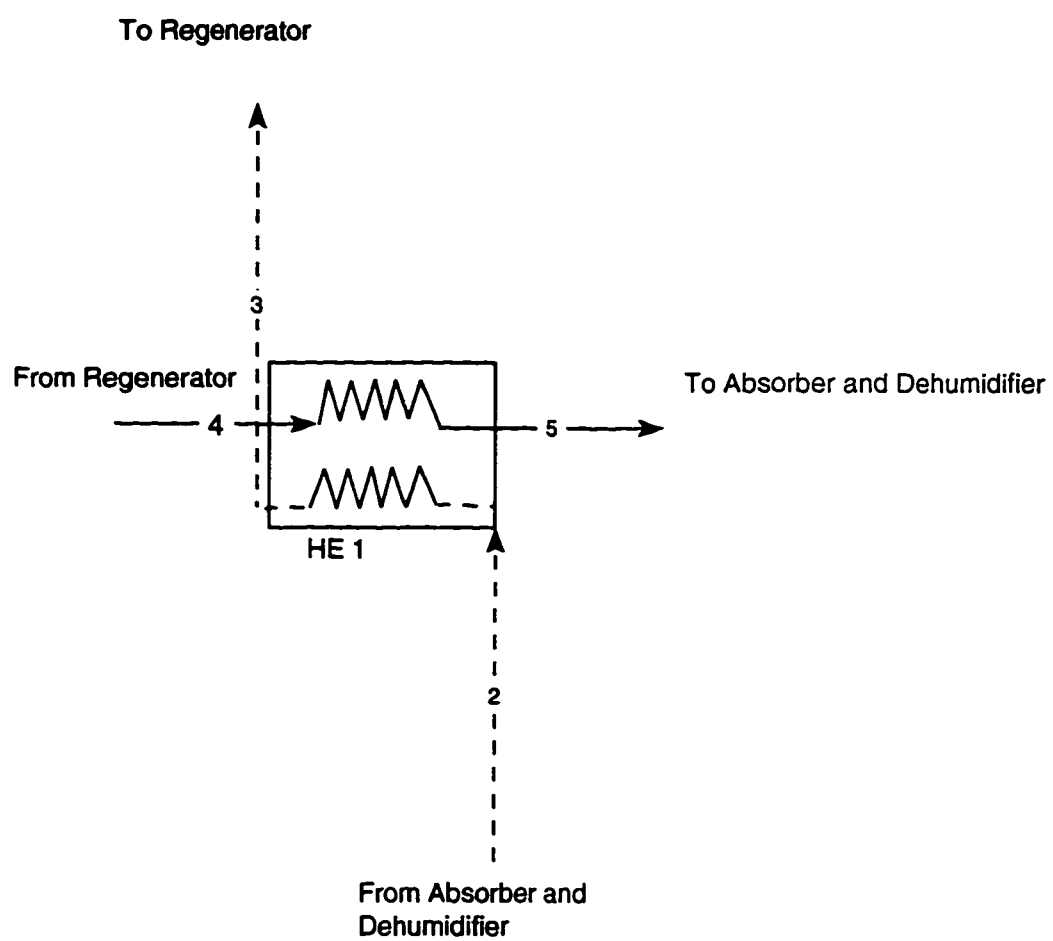


Figure 4.8: Hybrid Cycle Component: Heat Exchanger 1

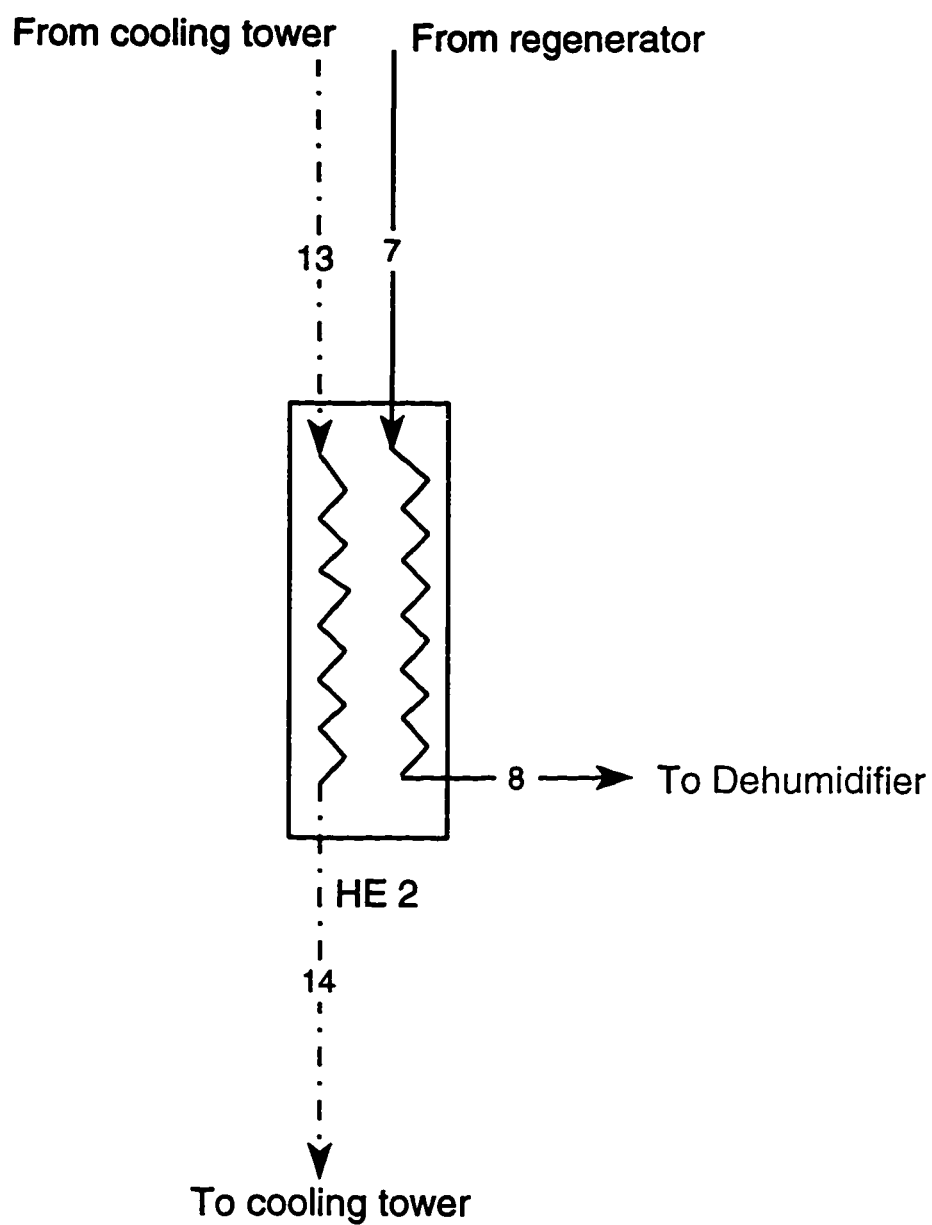


Figure 4.9: Hybrid Cycle Component: Heat Exchanger 2

Table 4.1: Hybrid Cycle Parameters

State pts.	t, ° C	h, kJ/kg	s, kJ/kg K	mass flow rate kg/s
1	40	122	1.6977	6
2	38	122	1.6341	11.2
3	60	159	2.3337	11.2
4	70	189	2.6517	10
5	60	181	2.3337	5
6	60	181	2.3337	5
7	60	181	2.3337	5
8	35	137	1.5387	5
9	44	143	1.8249	5.2
10	30	126	0.43	1
11	22	2541	8.6	1
12	42	175	0.58	1
13	30	126	0.43	1
14	38	159	0.55	1
15	28	117	0.35	1
16	22	92	0.22	1
17	30	126	0.43	1
18	35	146	0.5	1
rd,i	30	2556	8.355	0.06
rd,o	32	2570	8.255	0.03

Table 4.1, and substituting the values in the above equations yields the following quantities per unit mass flow rate of vapor released through the regenerator.

$$Q_{gen} = 2100 \text{ kW},$$

$$Q_{ab} = 2280 \text{ kW},$$

$$Q_{ev} = 2450 \text{ kW},$$

$$Q_{rd} = 420 \text{ kW},$$

$$Q_{gl} = 2300 \text{ kW},$$

Hence, the integrated vapor absorption and liquid desiccant system COP is es-

timated as,

$$COP = \frac{Q_{ev}}{Q_{gen}} = 1.166.$$

The hybrid vapor absorption and liquid desiccant system would produce a cooling effect of 700 tons of refrigeration at an evaporator temperature of 22⁰C per unit mass flow rate of the refrigerant. The COP obtained for the hybrid system is about 50 percent higher than the COP for the conventional absorption machine for the same conditions.

4.4.2 Dehumidifier Analysis

The dehumidification process can be accomplished in an equipment such as a finned-tube surface with column spray tower or packed tower. Since the packed tower facilitates high mass transfer by providing a large surface area in a relatively small volume, for the present study a packed bed dehumidifier is chosen. 25.4 mm ceramic Raschig rings are used as the packing material for the simulation study. The heat and mass transfer coefficients for the LiBr-air contacting system in a packed tower dehumidifier have been calculated as given in [23].

In order to predict the dehumidifier performance, a steady state model developed in [21], is used.

Referring to the differential element of the dehumidifier as shown in Fig. 4.10, a mass balance on the total control volume of the liquid and air gives,

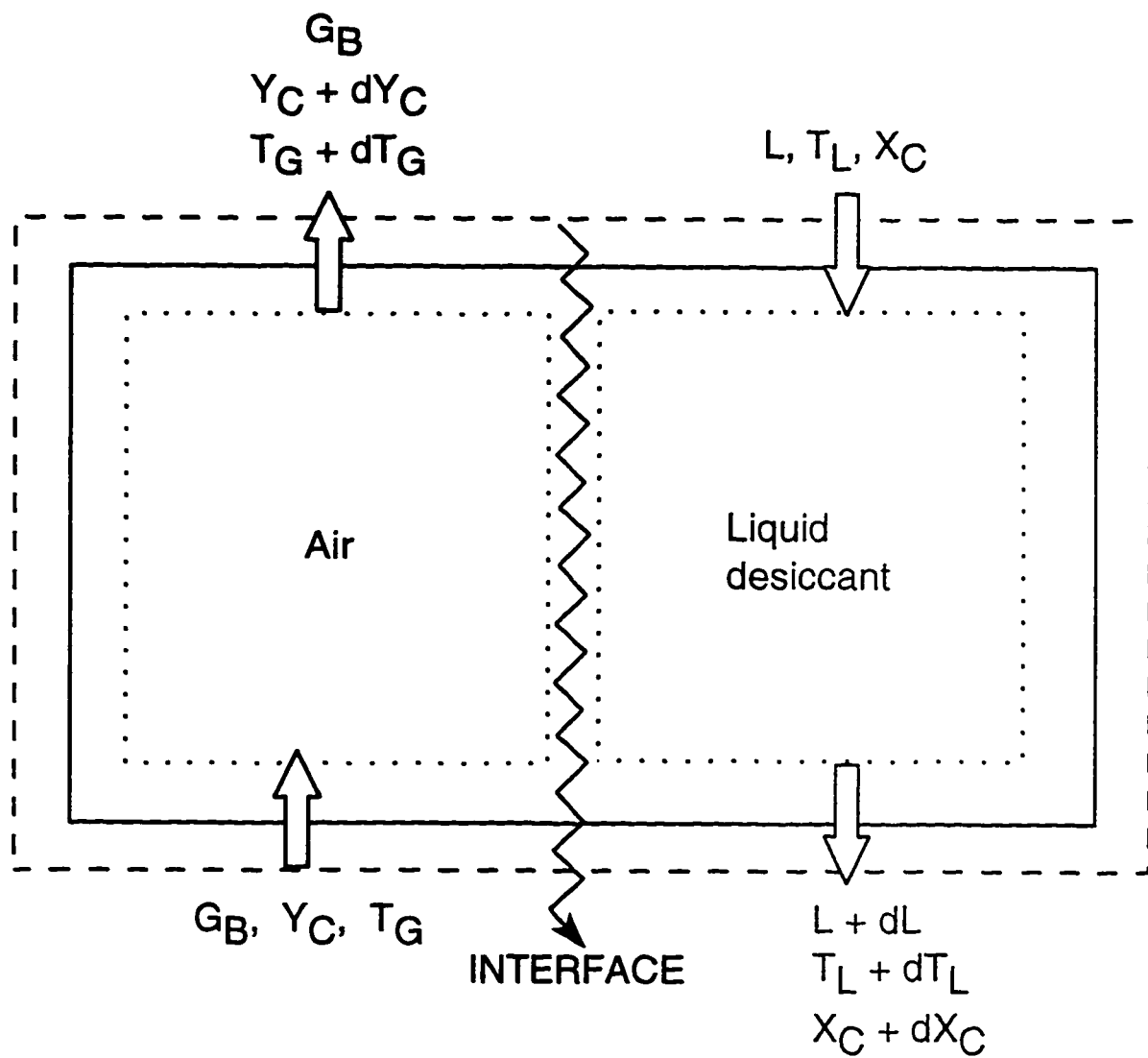


Figure 4.10: Differential Element of the Dehumidifier.

$$dL = GM_B dY_c \quad (4.11)$$

Taking the direction of mass and heat fluxes positive in the direction of air to liquid, the transfer of moisture from the air to the interface is given by [45]

$$N_c adZ = F_G \left[\ln \frac{1 - y_{cl}}{1 - y_c} \right] adz = -GM_B dY_c$$

or

$$\frac{dY_c}{dZ} = -\frac{F_G a}{GM_B} \left[\ln \frac{1 - y_{cl}}{1 - y_c} \right] \quad (4.12)$$

and heat transfer from air to the interface is,

$$q_G adZ = h'_G a (T_G - T_I) dZ \quad (4.13)$$

Now applying the Ackermann correction for simultaneous heat and mass transfer for air yields [20],

$$h'_G = -\frac{GM_B [C_c \frac{dY_c}{dZ}]}{1 - \exp[GM_B (C_c \frac{dY_c}{dZ})/h_G a]} \quad (4.14)$$

and for liquid

$$N_c adZ = F_L \ln \left[\frac{1 - x_c}{1 - x_{cl}} \right] adZ. \quad (4.15)$$

An energy balance on the air side gives,

$$GM_B H_G - [GM_B(H_G + dH_G) - GM_B dY_C [C_C(T_G - T_o) + \lambda T_o]] = h'_G a(T_G - T_I) dZ \quad (4.16)$$

where the air enthalpy per mole of dry air is

$$H_G = C_B(T_G - T_0) + Y_C [C_C(T_G - T_0) + \lambda T_0] \quad (4.17)$$

and

$$dH_G = C_B dT_G + Y_C C_C dT_G + dY_C [C_C(T_G - T_0) + \lambda T_0] \quad (4.18)$$

From eqns. 4.17 and 4.18, we obtain

$$h'_G a(T_G - T_I) dZ = -GM_B (C_B + Y_C C_C) dT_G$$

or

$$\frac{dT_G}{dZ} = -\frac{h'_G a(T_G - T_I)}{GM_B (C_B + Y_C C_C)} \quad (4.19)$$

The variation of ΔH_S with x_C is assumed to be negligible. Therefore the liquid temperature gradient and consequently the desiccant temperature at the outlet of the dehumidifier T_9 is given by the expression,

$$\frac{dT_L}{dZ} = \frac{GM_B}{LC_L} \left[(C_B + Y_C C_C) \frac{dT_G}{dZ} + [C_C(T_G - T_0) + \lambda T_0] \frac{dY_C}{dZ} - [C_L(T_L - T_0) + \Delta H_S] \frac{dY_C}{dZ} \right] \quad (4.20)$$

Equating Equations 4.13, 4.16 the interfacial conditions are obtained which turns out to be,

$$y_{cI} = 1 - (1 - Y_C) \left[\frac{1 - x_C}{1 - x_{CI}} \right]^{\frac{F_L}{F_G}} \quad (4.21)$$

Equation 4.21 is solved simultaneously with the vapor-liquid equilibrium data of lithium bromide solution, and the relation is found to be [21],

$$y_{cI} = 0.063647 + 0.001443t_I - \frac{0.04993}{x_{cI}} \quad (4.22)$$

The interfacial conditions x_{cI} and y_{cI} are determined by solving equations 4.21 and 4.22. Equations 4.13, 4.20 and 4.21 are used to calculate the successive steps of the tower height dZ beginning from the bottom of the tower $Z = 0$.

The dehumidifier model is solved numerically for a maximum humidity of 0.028 kg of water per kg of dry air and a minimum of 0.014 kg of water /kg of dry air. The maximum and minimum liquid temperatures considered are 35°C and 30°C respectively. The air flux is selected as 3 kg/s-m² per unit area of the dehumidifier cross section. Figure 4.11 shows the psychrometric chart for the process of dehumidification and sensible cooling. The mixture of return air from the room (state 'e') and fresh air from the outdoor (state 'a') enters the dehumidifier (state 'b') at 30 °C and 0.020 kg H₂O/kg dry air. The dry air at state 'c' is passed over the cooling coil where it is sensibly cooled and enters the room at state 'd'. The conditioned air picks up the load in the room and leaves at state 'e'.

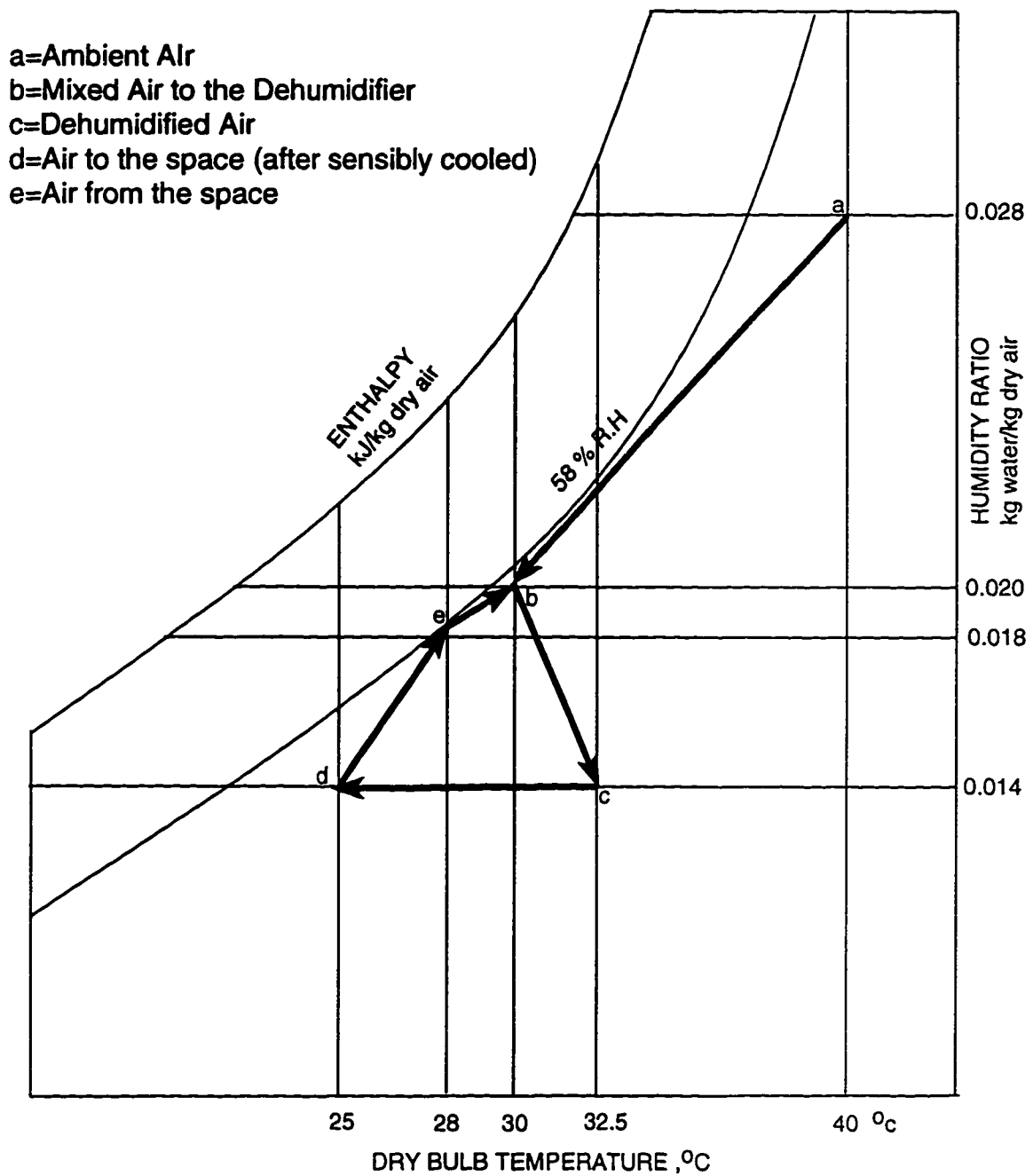


Figure 4.11: Hybrid Cycle Processes Shown on a Psychrometric Chart.

Chapter 5

EXERGY ANALYSIS

5.1 Introduction

Exergy is defined as the maximum work which could possibly be extracted from a thermodynamic system in a given state when it is brought in equilibrium with its environment. It can also be stated as the minimum amount of work which could possibly be expended to bring the system from a state of equilibrium with the environment to some desired state. Solar thermal radiation is rich in exergy which is likely to be tapped efficiently only with an indepth understanding of second law of thermodynamics. The second law can help us in finding ways to enhance the collection of exergy from solar insolation.

Most of the research work has been done in the field of exergy analysis of flat plate solar collectors which are typically used in closed cycle refrigeration and air-

conditioning systems. Little efforts have so far been devoted to open-cycle solar absorption refrigeration systems, where the absorbent-solution is directly exposed to the atmosphere and most of the interactions are with the ambient [13]. In this chapter, exergy analysis of a hybrid liquid desiccant and vapor absorption air-conditioning system has been studied. A partly closed-open solar collector has been used for regeneration of the weak (or dilute) desiccant solution. In this type of collector, the solution is preliminary heated in the closed (glazed) section where evaporation does not take place. Due to its increased temperature in the glazed section, the evaporation of the moisture in the open section of the collector is intensified.

5.2 Regenerator Analysis

In this section, we present first-law-based thermodynamic analysis of the regenerator of the hybrid cycle.

5.2.1 Glazed Section

It is assumed that the temperature, concentration, and vapor pressure of the desiccant are constant. The properties of these parameters are taken at the arithmetic average values at the beginning and end of each section of the regenerator. Referring to Fig 5.1, energy balance equation for the glazed section of the regenerator can be

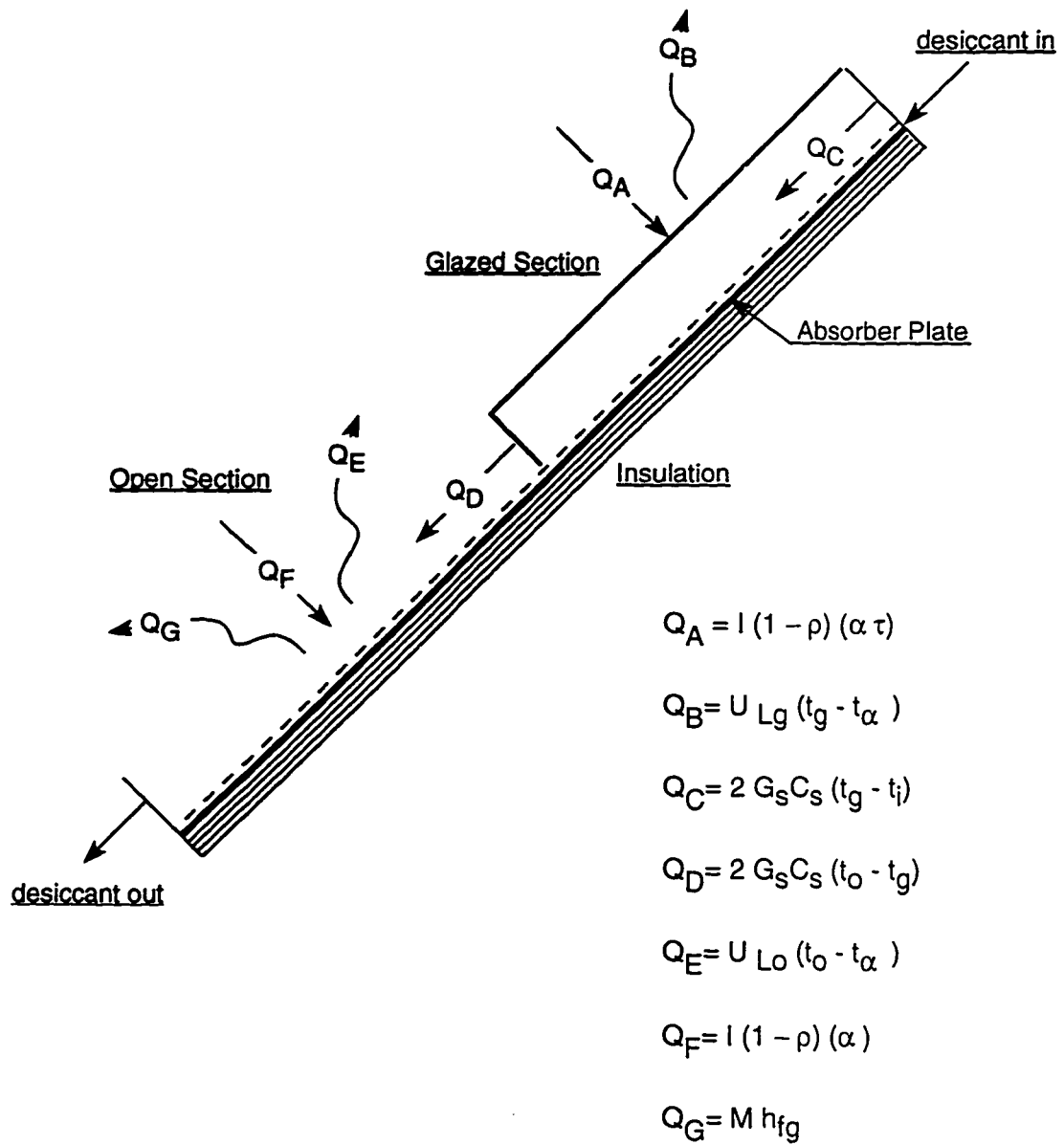


Figure 5.1: Exergy Analysis : Partly Closed-Open Solar Regenerator.

written as [46]

$$I(1 - \rho)(\tau\alpha)A_g - U_{L,g}A_g(t_g - t_\infty) - 2G_sC_s(t_g - t_i) = 0 \quad (5.1)$$

which gives the desiccant temperature in the glazed section as,

$$t_g = \frac{I(1 - \rho)(\tau\alpha)A_g + U_{L,g}A_g t_\infty + 2G_sC_s t_i}{U_{L,g}A_g + 2G_sC_s} \quad (5.2)$$

and the useful heat transfer in the glazed section of the regenerator can be written as ,

$$Q_g = IA_g - [U_{L,g}A_g(t_g - t_\infty) + 2G_sC_s(t_g - t_i)] \quad (5.3)$$

5.2.2 Open Section

Energy balance for the open section of the regenerator can be expressed as [46]

$$I(1 - \rho)(\alpha)A_o - U_{L,o}A_o(t_o - t_\infty) - 2G_sC_s(t_o - t_g) - Mh_{fg} = 0 \quad (5.4)$$

The useful heat transfer in the open section of the regenerator is,

$$Q_o = IA_o - [U_{L,o}A_o(t_o - t_\infty) + 2G_sC_s(t_o - t_g) + Mh_{fg}] \quad (5.5)$$

Therefore the desiccant temperature in the open section can be expressed as [46]

$$t_o = \frac{1}{b} \left[\frac{M}{\beta A_o} + P_\infty - a - \frac{c}{\xi} \left(1 - \frac{M}{2G_s} \right) \right] \quad (5.6)$$

where the mass of water evaporated from the open section of the regenerator is

$$M = \beta A_o (P_o - P_\infty) \quad (5.7)$$

The relationship between vapor pressure, temperature, and concentration of the desiccant can be expressed as [13]

$$P = a + bt + (c/\xi) \quad (5.8)$$

Simplification of equations 5.6, 5.7, and 5.8 yields the rate of moisture evaporation in the open section of the regenerator. It is found to be [46]

$$M = \frac{A_o[I(1 - \rho)(\alpha) + U_{L,o}t_\infty] + 2G_sC_s t_g + \frac{1}{b}(a + \frac{c}{\xi} - P_\infty)(U_{L,o}A_o + 2G_sC_s)}{\frac{1}{b}(\frac{1}{\beta A_o} + \frac{c}{2\xi G_s})(U_oA_o + 2G_sC_s) + h_{fg}} \quad (5.9)$$

Lithium Bromide has been chosen as the liquid desiccant in the present study. The empirical constants used in the above expressions are $a = -78$, $b = 0.87$, $c = 2810$. The following values for the parameters considered in the equations were assumed: $U_{L,g} = 7.5W/m^2.^{\circ}C$, $U_{L,o} = 19W/m^2.^{\circ}C$, $\alpha = 0.9$. and $\tau = 0.85$. Furthermore the ambient temperature was taken as $t_o = 35^{\circ}C$, and h_{fg} was assumed to be constant at 2550 kJ/kg.

5.3 Exergy Analysis

In ordinary flat plate collectors the energy of the fluid generated by adding pump work and passing through the tubes is converted into thermal energy by fluid friction. The exergy loss in the process is zero if the thermal insulation is perfect, which is not

possible in practice, therefore the exergy of the collector usually decreases. However, in open cycle regenerators, the desiccant flows over the absorber plate and no tubes are used, and hence, the friction and pressure losses are therefore assumed to be negligible in the present study. However, convection, evaporation, and radiation losses have been taken into consideration in the analysis of the system.

5.3.1 Glazed Section

The useful exergy in the glazed section is due to heat transfer and fluid flow. This can be expressed as [47]

$$(E_{in} - E_{out})_g = Q_g(1 - \frac{T_\infty}{T_g}) + G_s C_s [T_g - T_i - T_\infty \ln(\frac{T_g}{T_i})] \quad (5.10)$$

5.3.2 Open Section

The flow of exergy in the open section is due to heat transfer, mass flow rate and exergy rate and exergy lost due to the mass transfer to the ambient [47]

$$(E_{in} - E_{out})_o = Q_o(1 - \frac{T_\infty}{T_o}) + G_s C_s [T_o - T_g - T_\infty \ln(\frac{T_o}{T_g})] - Mm \quad (5.11)$$

where Mm is the exergy lost due to the moisture evaporation and is given by

$$Mm = MC_v [T_\infty - T_o - T_\infty \ln(\frac{T_\infty}{T_o})] - \frac{\bar{R}}{MW} T_\infty \ln(\frac{x_v P_\infty}{P_v}) \quad (5.12)$$

The irreversibility rate or the exergy destruction is therefore represented as,

$$\dot{I} = (E_{in} - E_{out})_g + (E_{in} - E_{out})_o \quad (5.13)$$

For each component of the Hybrid cycle shown in Figs 5.2, 5.3, 5.4 and 5.5, the energy and exergy change can be calculated as follows [47] and [48]

5.3.3 Evaporator

The evaporator absorbs heat from the cooling coil during the process of evaporation by an amount,

$$Q_{ev} = \dot{m}_{10}(h_{11} - h_{10}) = \dot{m}_{16}(h_{16} - h_{15}) \quad (5.14)$$

It is assumed that the evaporator absorbs heat from the cooling coil at a constant evaporator temperature T_{ev} . The irreversible losses in the expansion valve is therefore given by the expression,

$$\dot{I}_{ev} = \dot{m}_{11}(e_{11} - e_{10}) - Q_{ev}\left(1 - \frac{T_{\infty}}{T_{ev}}\right) \quad (5.15)$$

where

$$(e_{11} - e_{10}) = h_{11} - h_{10} - T_{\infty}(s_{11} - s_{10}) \quad (5.16)$$

A similar analysis is performed to different components of the hybrid cycle.

5.3.4 Absorber

The exergy and irreversibility expressions in the absorber are given by,

$$Q_{ab} = \dot{m}_{11}h_{11} + \dot{m}_6h_6 - \dot{m}_1h_1 = \dot{m}_{17}(h_{17} - h_{18}) \quad (5.17)$$

$$\dot{I}_{ab} = \dot{m}_{11}e_{11} + \dot{m}_6e_6 - \dot{m}_1e_1 - Q_{ab}\left(1 - \frac{T_\infty}{T_{ab}}\right) \quad (5.18)$$

$$e_{11} = h_{11} - h_\infty - T_\infty(s_{11} - s_\infty) \quad (5.19)$$

$$e_6 = h_6 - h_\infty - T_\infty(s_6 - s_\infty) \quad (5.20)$$

$$e_1 = h_1 - h_\infty - T_\infty(s_1 - s_\infty) \quad (5.21)$$

5.3.5 Heat Exchanger 1

The exergy and irreversibility expressions in the heat exchanger 1 are given by,

$$\dot{I}_{HE1} = \dot{m}_2e_2 + \dot{m}_4e_4 - \dot{m}_3e_3 - \dot{m}_5e_5 \quad (5.22)$$

$$e_2 = h_2 - h_\infty - T_\infty(s_2 - s_\infty) \quad (5.23)$$

$$e_4 = h_t - h_\infty - T_\infty(s_4 - s_\infty) \quad (5.24)$$

$$e_3 = h_3 - h_\infty - T_\infty(s_3 - s_\infty) \quad (5.25)$$

$$e_5 = h_5 - h_\infty - T_\infty(s_5 - s_\infty) \quad (5.26)$$

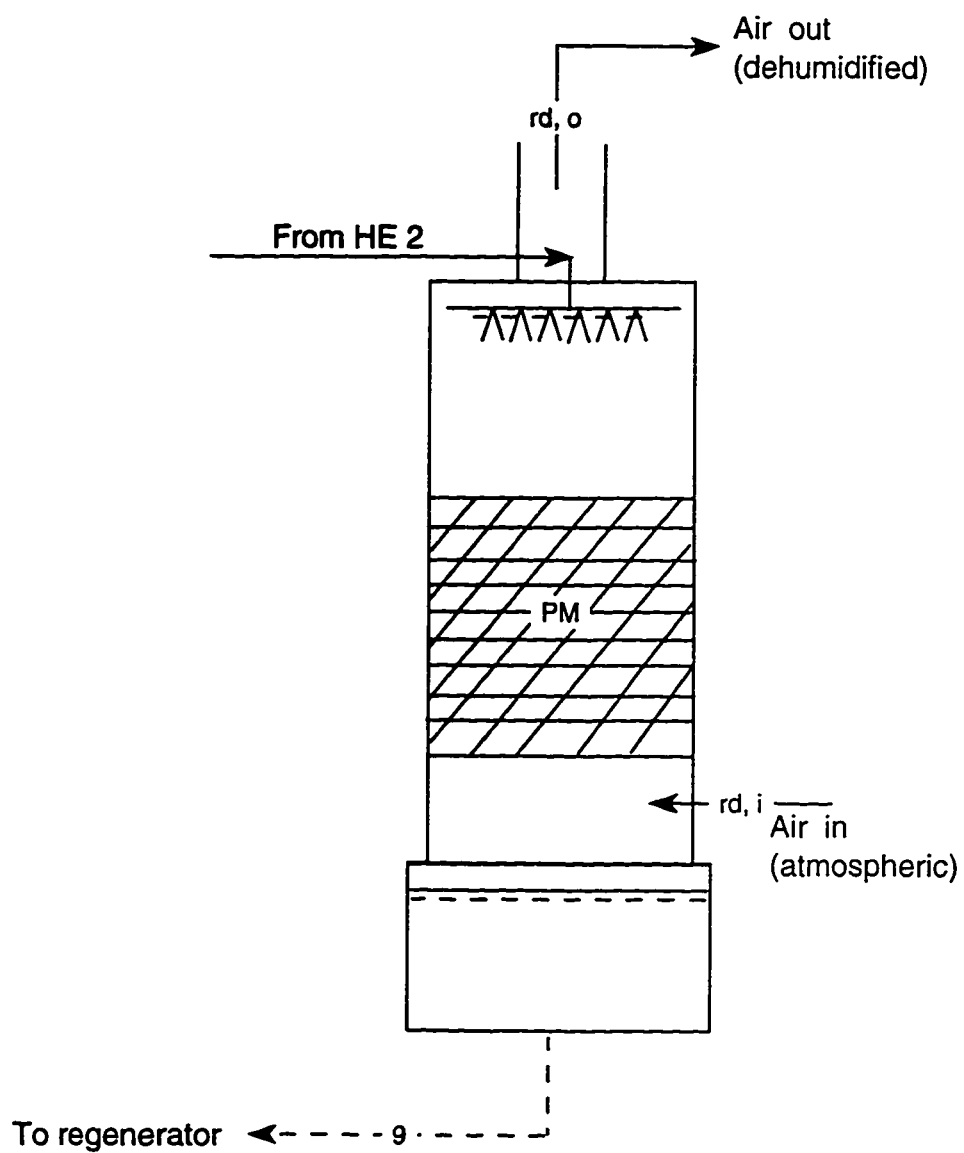


Figure 5.2: Dehumidifier

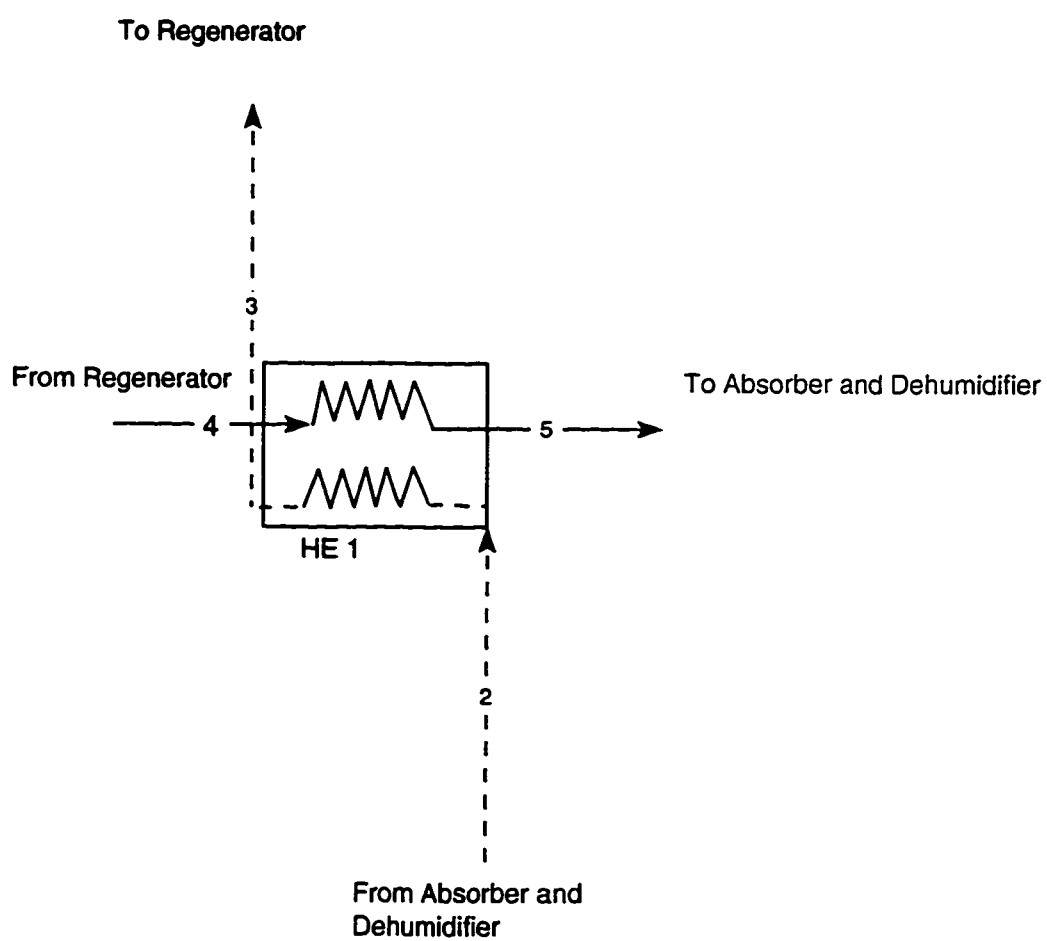


Figure 5.3: Heat Exchanger 1

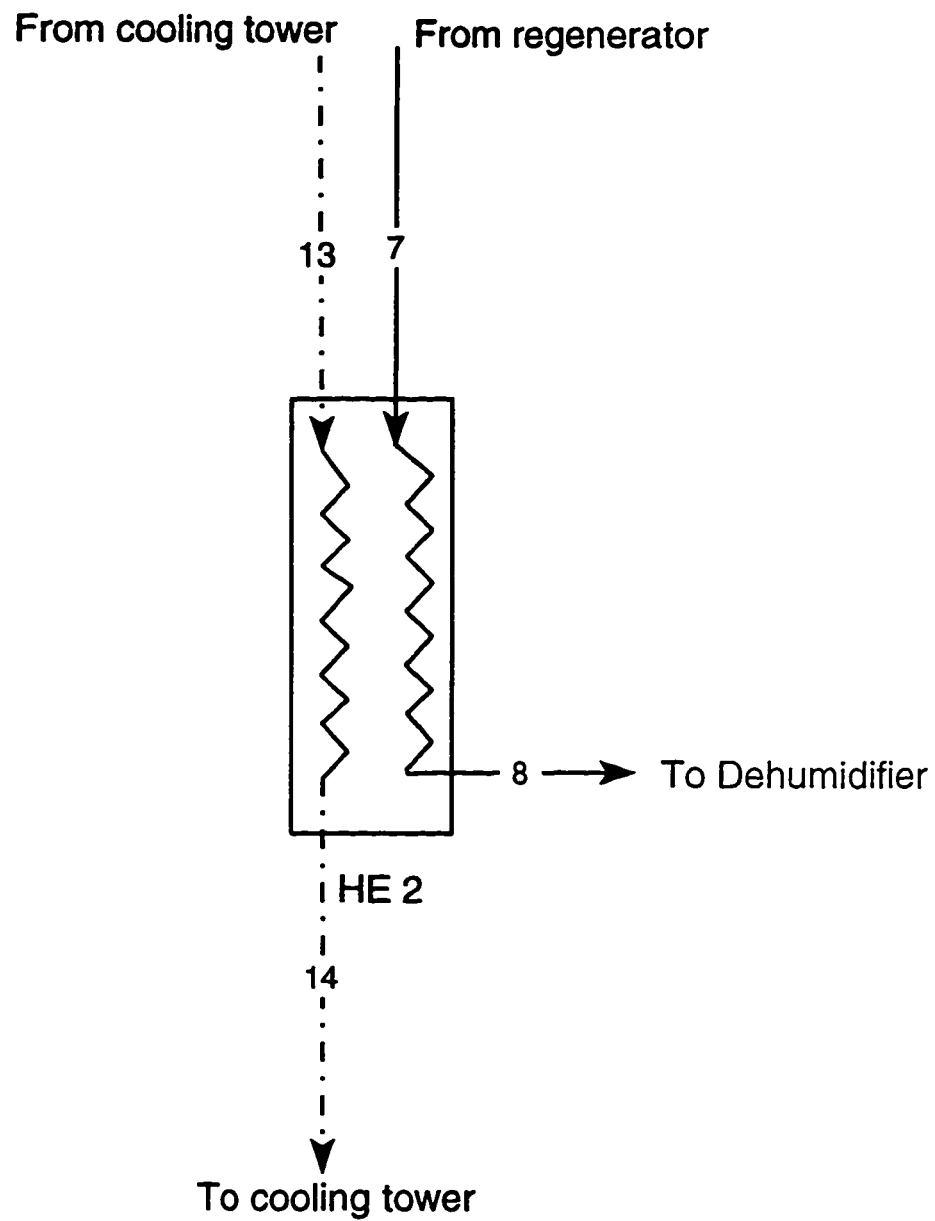


Figure 5.4: Heat Exchanger 2

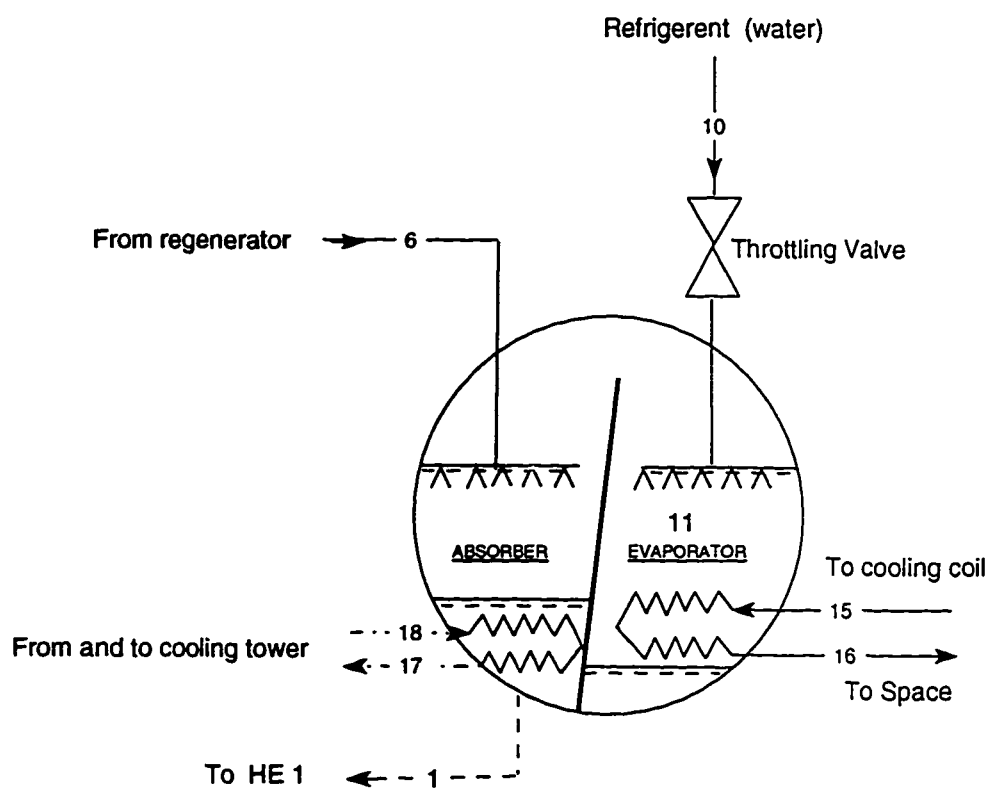


Figure 5.5: Absorber and Evaporator

5.3.6 Heat Exchanger 2

The exergy and irreversibility expressions in the heat exchanger 2 are given by,

$$\dot{I}_{HE2} = \dot{m}_{13}e_{13} + \dot{m}_7e_7 - \dot{m}_{14}e_{14} - \dot{m}_8e_8 \quad (5.27)$$

$$e_{13} = h_{13} - h_\infty - T_\infty(s_{13} - s_\infty) \quad (5.28)$$

$$e_7 = h_7 - h_\infty - T_\infty(s_7 - s_\infty) \quad (5.29)$$

$$e_{14} = h_{14} - h_\infty - T_\infty(s_{14} - s_\infty) \quad (5.30)$$

$$e_8 = h_8 - h_\infty - T_\infty(s_8 - s_\infty) \quad (5.31)$$

5.3.7 Throttling valve

The exergy and irreversibility expressions in the throttling valve are given by,

$$\dot{I}_{thr} = \dot{m}_{10}(e_{10} - e_{11}) \quad (5.32)$$

$$e_{10} = h_{10} - h_\infty - T_\infty(s_{10} - s_\infty) \quad (5.33)$$

$$e_{11} = h_{11} - h_\infty - T_\infty(s_{11} - s_\infty) \quad (5.34)$$

5.3.8 Dehumidifier

The exergy and irreversibility expressions in the dehumidifier are given by,

$$Q_{rd} = \dot{m}_8 h_8 + \dot{m}_{rd,i} h_{rd,i} - \dot{m}_9 h_9 - \dot{m}_{rd,o} h_{rd,o} = \dot{m}_{14} (h_{14} - h_{12}) \quad (5.35)$$

$$\dot{I}_{rd} = \dot{m}_8 e_8 + \dot{m}_{rd,i} e_{rd,i} - \dot{m}_9 e_9 - \dot{m}_{rd,o} e_{rd,o} - Q_{rd} \left(1 - \frac{T_\infty}{T_{rd}}\right) \quad (5.36)$$

$$e_8 = h_8 - h_\infty - T_\infty (s_8 - s_\infty) \quad (5.37)$$

$$e_9 = h_9 - h_\infty - T_\infty (s_9 - s_\infty) \quad (5.38)$$

$$e_{rd,i} = h_{rd,i} - h_\infty - T_\infty (s_{rd,i} - s_\infty) \quad (5.39)$$

$$e_{rd,o} = h_{rd,o} - h_\infty - T_\infty (s_{rd,o} - s_\infty) \quad (5.40)$$

Referring to Table 5.1 and performing the calculation as per the above expressions we get the irreversibility rate in for evaporator, absorber, heat exchanger 1, heat exchanger 2, throttling valve, and the dehumidifier.

Table 5.1: Exergy Analysis:Component Parameters

State pts.	t, ° C	h, kJ/kg	s, kJ/kg K
1	40	122	1.6977
2	38	122	1.6341
3	60	159	2.3337
4	70	189	2.6517
5	60	181	2.3337
6	60	181	2.3337
7	60	181	2.3337
8	35	137	1.5387
9	44	143	1.8249
10	30	126	0.43
11	22	2541	8.6
12	42	175	0.58
13	30	126	0.43
14	38	159	0.55
15	28	117	0.35
16	22	92	0.22
17	30	126	0.43
18	35	146	0.5
rd,i	40	2575	8.255
rd,o	30	2556	8.455

Chapter 6

EXPERIMENTAL STUDY OF SOLAR REGENERATORS

6.1 Introduction

The open cycle absorption solar cooling system provides an alternative to the conventional cooling system for, not only energy saving but also for using refrigerants that have no detrimental effects on the environment. The hybrid system is operated by pumping the weak absorbent (desiccant) solution from the outlet of the absorber and the dehumidifier into the open solar regenerator. The water-rich, weak desiccant solution flows as a thin falling film on the inclined black surface of the regenerator where it absorbs the solar energy and releases refrigerant (water) vapor into the ambient air. Then the regenerated strong solution can be fed back to the absorber

and the dehumidifier after sensibly cooled in a heat exchanger. The cooling effect of this hybrid system is provided by evaporation of the water (refrigerant) into vapor in the evaporator and the dehumidification is provided by the absorption of water vapor in the dehumidifier. The vapor is then removed by the strong solution via an absorption process taking place in the absorber and the dehumidifier to complete the cycle. The energy required for the regeneration process comes primarily from the sun. The rate of water evaporated from the regenerator gives the direct measurement of the system cooling and dehumidification capacity. Therefore the regenerator is the key component of the system and its performance is of major concern in the development of the hybrid cycle absorption systems.

6.2 Types of Solar Regenerators

The four different types of solar regenerators that have been experimentally studied are,

1. Fully Open Regenerator
2. Fully Closed Regenerator
3. Partly Closed-Open Regenerator
4. Alternately Closed-Open Regenerator

6.2.1 Fully Open Regenerator

A fully open type of regenerator is entirely open to the atmosphere and is in direct contact with the ambient as shown in Fig. 6.1. As the weak/dilute desiccant solution flows over the collector, it loses its moisture to the atmosphere. Most of the moisture is carried away by the atmospheric air as soon as it is liberated by the weak desiccant solution thereby creating a lower vapor pressure zone for further evaporation of the moisture. This regenerator has the disadvantage of being contaminated with dust and particles which may affect the solution properties as well as the performance of the regenerator. Also in high humid climate, this type of regenerator is not that efficient because of the tendency of high ambient vapor pressure to reduce the rate of moisture evaporation. Therefore in such situations a preincorporated glazed section at beginning of the regenerator would be necessary to increase the temperature of the incoming weak desiccant and thereby enhances the overall rate of moisture evaporation in the open section. The thermal network for fully open solar regenerator is shown in Fig. 6.2.

6.2.2 Fully Closed Regenerator

The fully closed type of regenerator consists of a glass plate covered over the regenerator as shown in Fig. 6.3. As the solution flows over the collector, the moisture gets evaporated and condenses under the glass plate. This moisture is collected

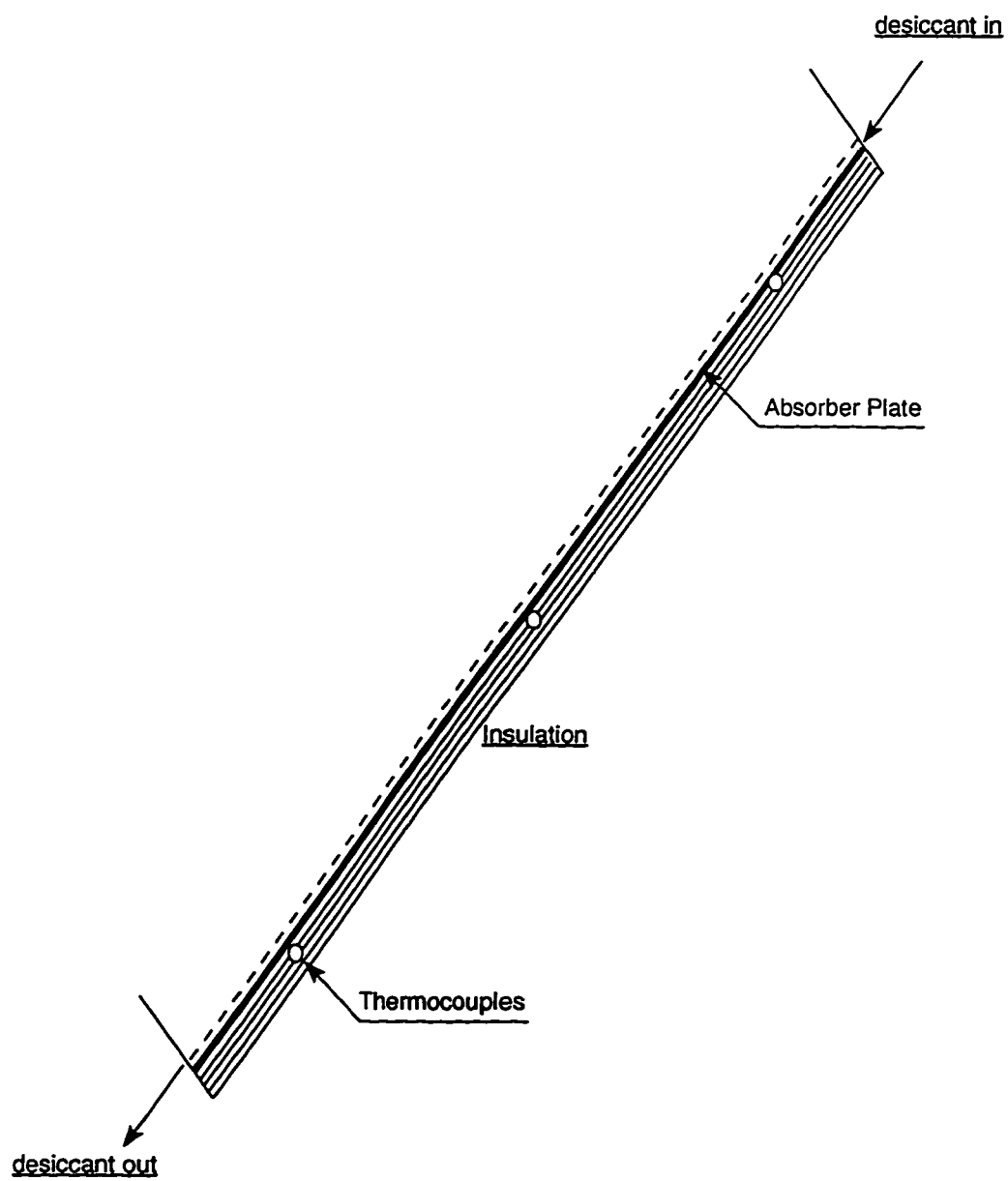


Figure 6.1: Fully Open Solar Regenerator.

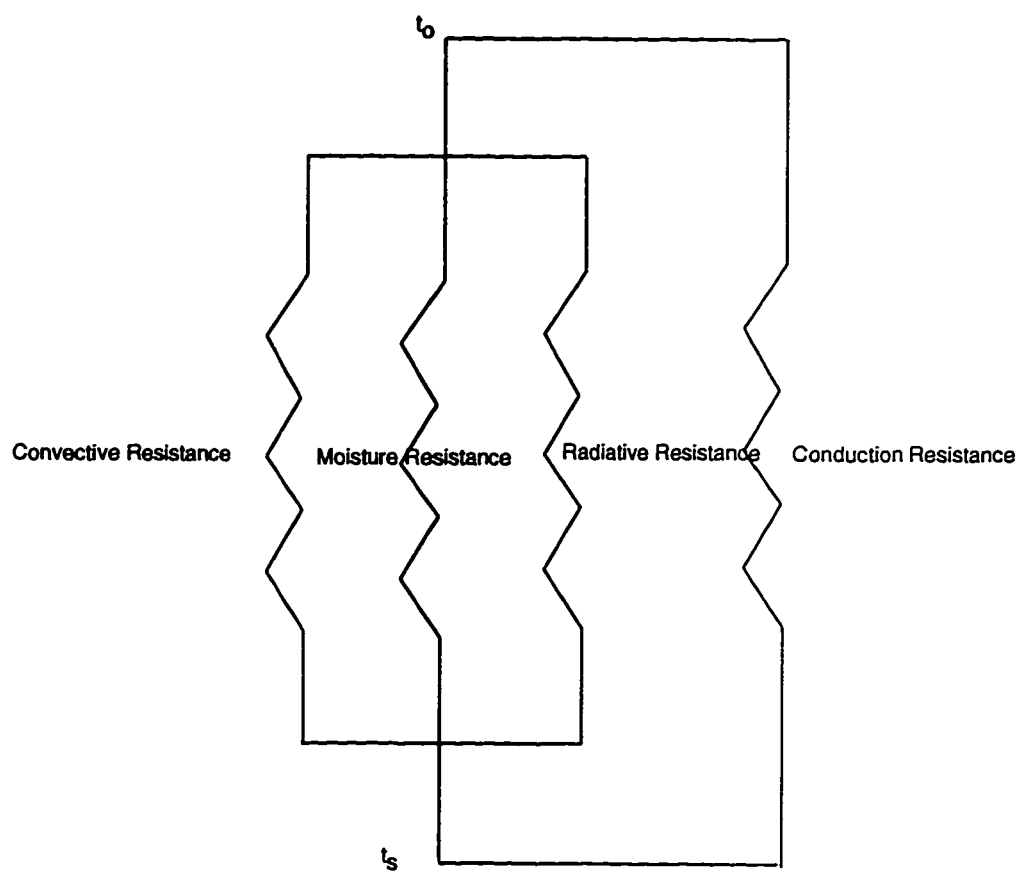


Figure 6.2: Thermal Network for Fully Open Solar Regenerator.

separately and the rate of moisture evaporation is calculated. Also, as the moisture evaporated gets condensed under the glass plate, it forms a thin film below the glass plate, which in turn creates a barrier for insolation and high rates of moisture evaporation. The rate of moisture evaporation depends on the temperature of the desiccant solution and its corresponding vapor pressure. Since the glass cover plate is at higher temperature when compared to the ambient temperature, the condensate under it is also at a higher temperature. The vapor pressure corresponding to the glass cover temperature is also likely to be higher than that of ambient. This results in decrease in the rate of moisture evaporation in the fully closed regenerator due to low vapor pressure difference between the desiccant solution and the condensate. There is a specific height of the glass cover plate from the absorber surface at which maximum rate of moisture evaporation occur. The thermal network for fully closed solar regenerator is shown in Fig. 6.4.

6.2.3 Partly Closed-Open Regenerator

The partly closed/open type of solar regenerator is mostly applied in high humid regions where the ambient vapor pressures are high. Upper part of this regenerator is glazed and lower part of it is open as shown in Fig. 6.5. The purpose of the glazed section is to initially increase the temperature of the incoming dilute desiccant solution. As the solution temperature increases, its vapor pressure increases and hence, its ability to release moisture in the open section of the regenerator increases.

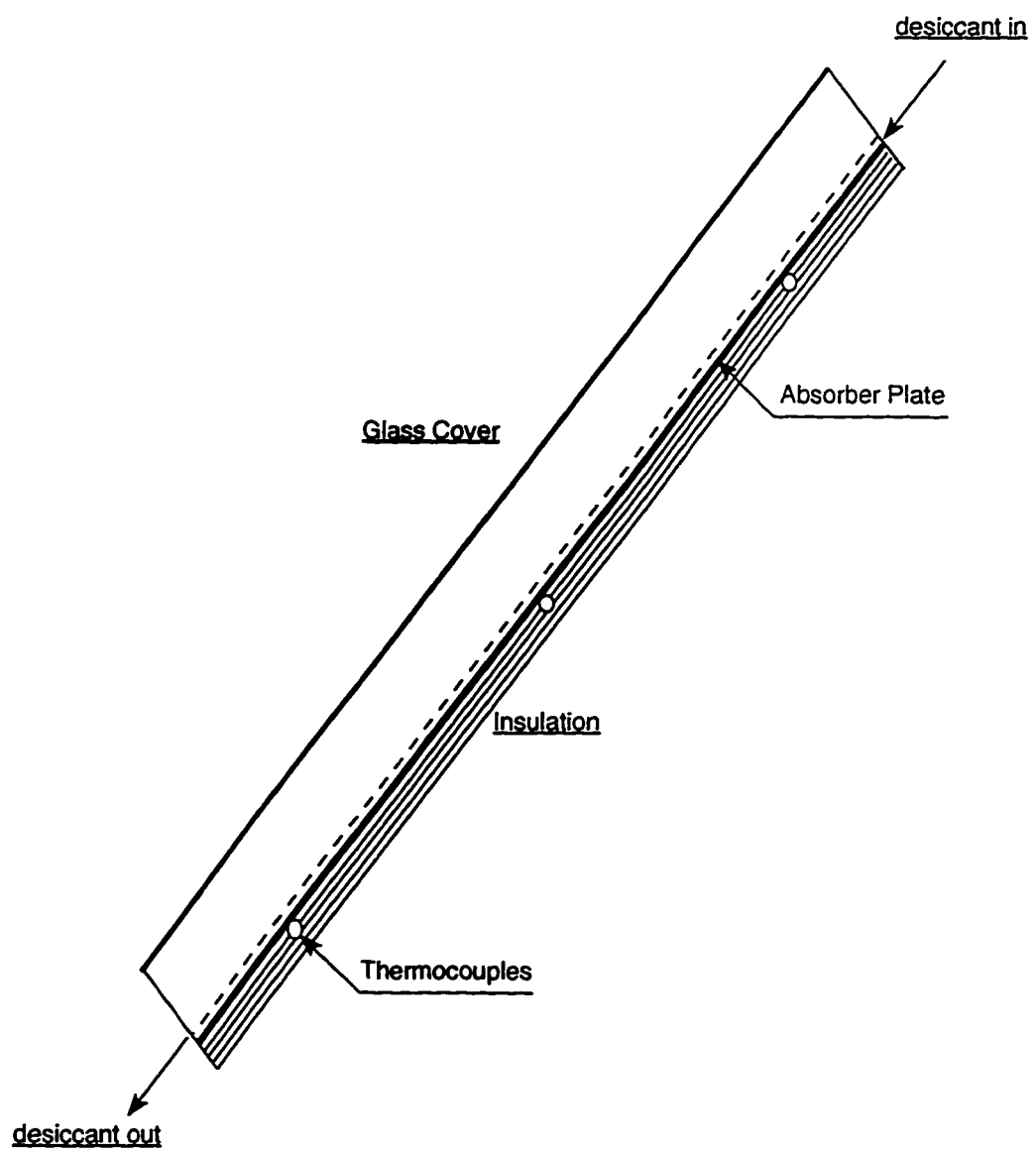


Figure 6.3: Fully Closed Solar Regenerator.

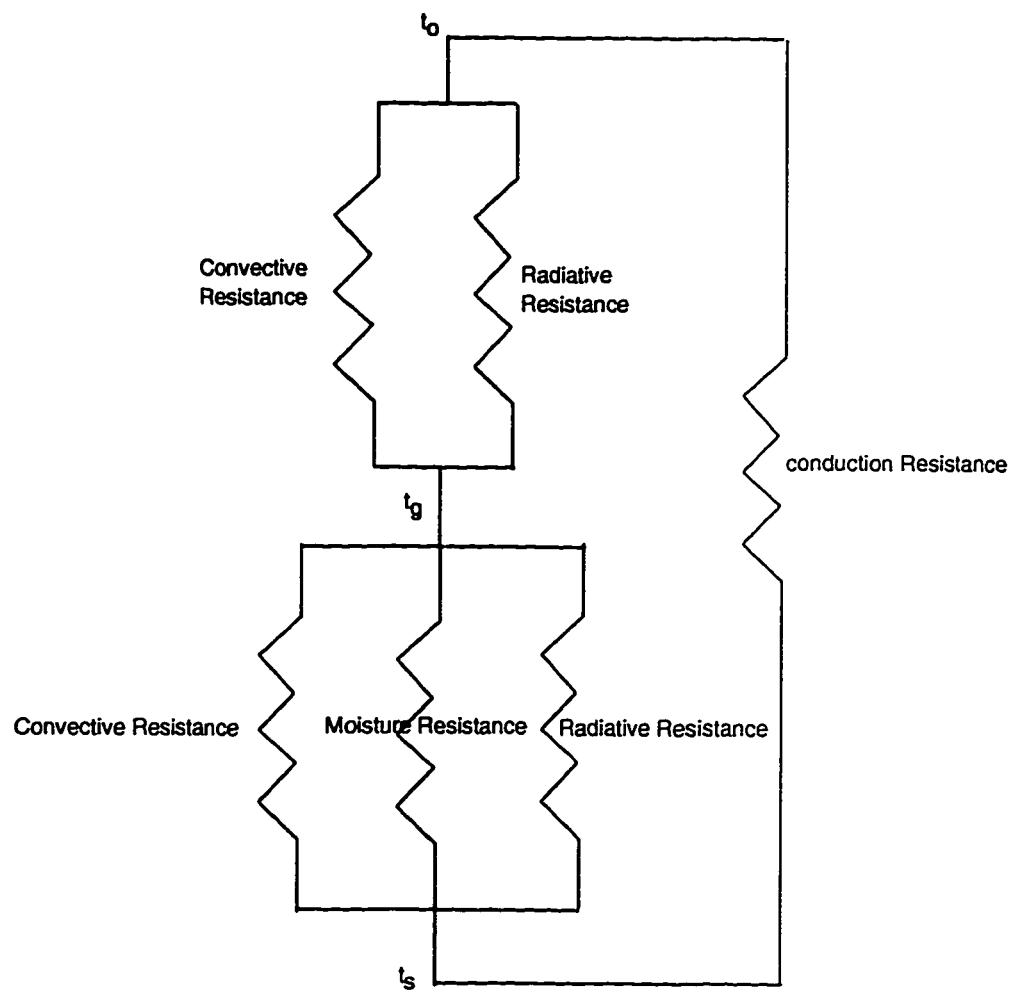


Figure 6.4: Thermal Network for Fully Closed Solar Regenerator.

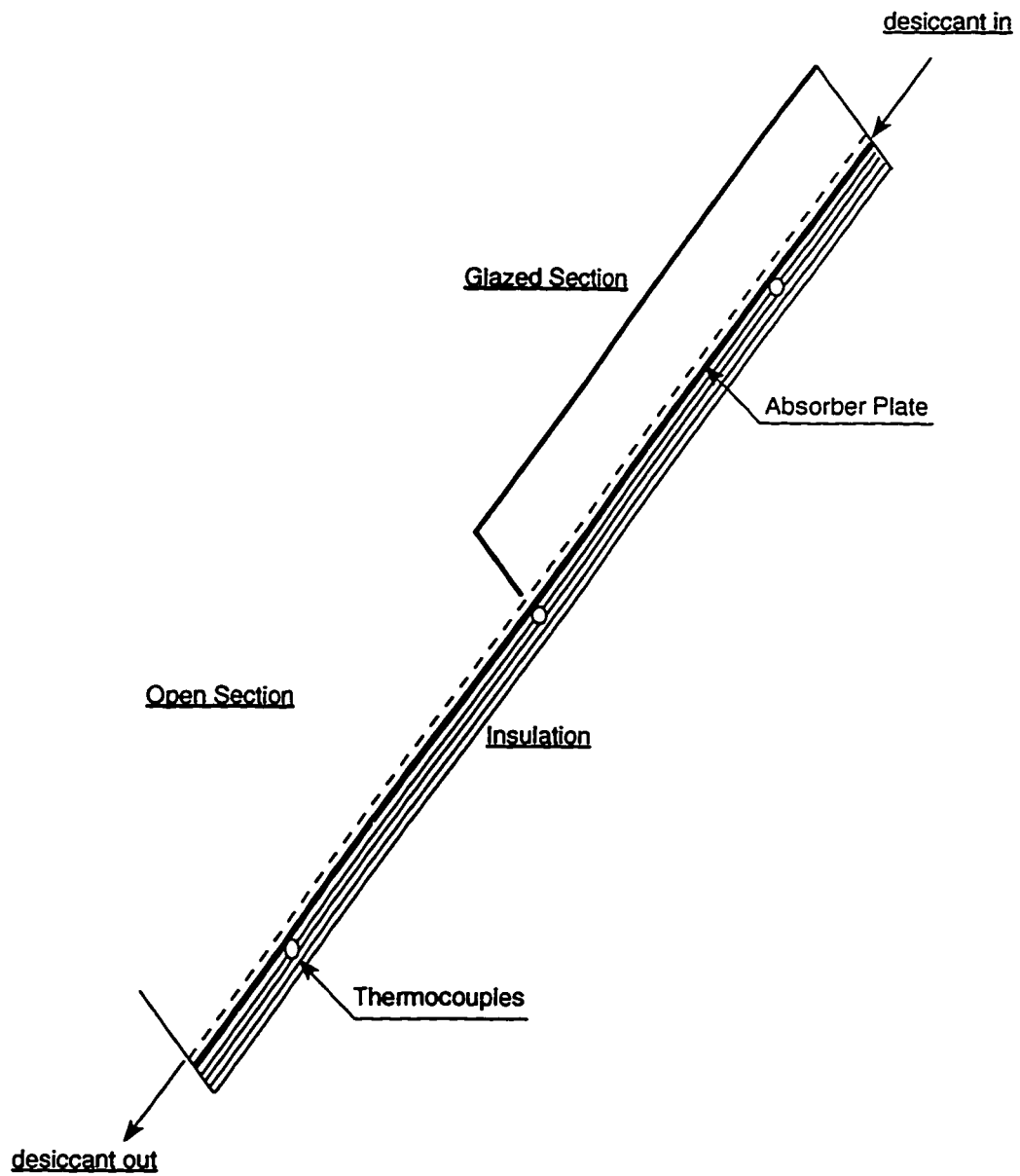


Figure 6.5: Partly Closed-Open Solar Regenerator.

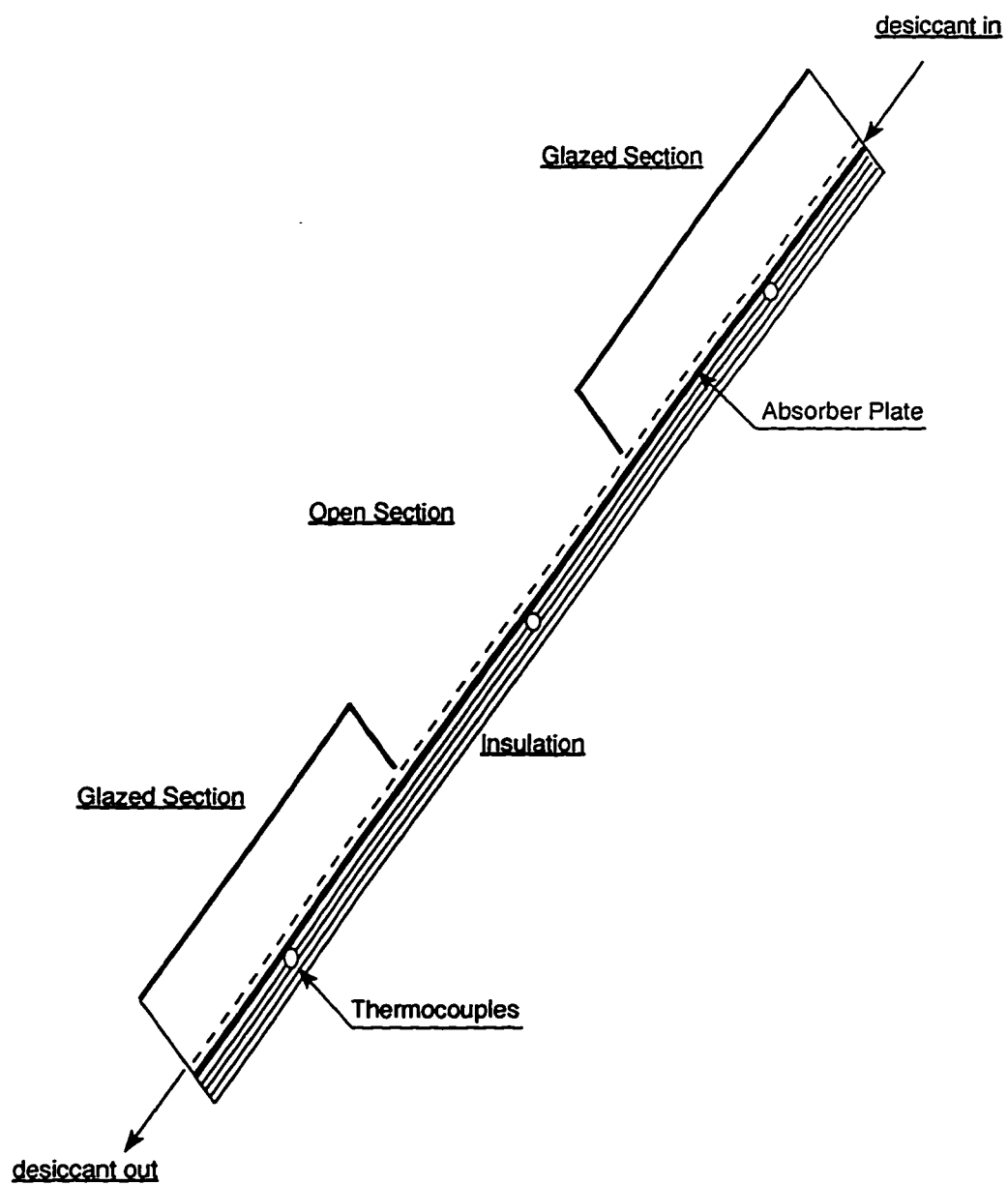


Figure 6.6: Alternately Closed-Open Solar Regenerator.

Therefore, the solution at the outlet of glazed section in the partly closed/open type of regenerator is both at high temperature as well as high vapor pressure.

6.2.4 Alternately Closed-Open Regenerator

An alternately closed-open regenerator consists of alternately glazed and open sections as shown in Fig. 6.6. In this regenerator, every glazed section increases the temperature of the incoming weak/dilute desiccant solution and every open section evaporates the moisture. Hence, as the desiccant passes from a glazed section to an open section, its rate of moisture evaporation increases. The next glazed section further increases its temperature, thereby further increasing its ability to evaporate the moisture. The net effect is a desiccant with high temperature and concentration.

6.3 Performance Evaluation

Performance evaluation of glazed and unglazed solar collectors was done by Hawlader et al [18]. It was found that the performance of unglazed solar regenerator was better than that of a glazed regenerator. This performance was evaluated in terms of quantity of moisture evaporated from the regenerator. A report on experimental study of the heat and mass transfer in solar collector/regenerator for open-cycle liquid desiccant regeneration was prepared by Ji and Wood [19]. An experimental analysis was performed by Yang and Wang [49] to study the effect of a heat ex-

changer for recovering the heat on the performance of a glazed solar collector. The test results showed that there was an increase of about 25% in the performance of the collector. Gandhidasan [50] presented a testing procedure for determining the performance of an open solar regenerator. The effect of different meteorological and solution parameters on the performance of the regenerator was also studied.

The instantaneous efficiency of the regenerator is defined as the energy used in the mass transfer (moisture evaporation) over some specified time period to the incident solar energy over the same time period. The performance of the regenerator depends on various parameters such as desiccant flow rate, operating temperature, concentration and temperature of the desiccant at the inlet of the regenerator, insolation, wind speed conditions, outdoor temperature and humidity, etc. The concentration of the desiccant at the inlet of the regenerator was maintained constant. At inlet the mass flow rate of the desiccant solution was measured by a rotameter. As the weak desiccant flows over the absorber plate its concentration increases from ξ_i to ξ_o due to evaporation of moisture. The mass of moisture evaporated from the desiccant was calculated by measuring the inlet and outlet concentrations of the desiccant.

The following equation was used to calculate the mass of moisture evaporated from the desiccant [50],

$$m = G_s \left(1 - \frac{\xi_i}{\xi_o}\right)$$

where,

m =mass of moisture evaporated, kg/h m².

G_s = mass flow rate of the desiccant at the inlet, kg/h m².

ξ_i = desiccant concentration at the inlet.

ξ_o = desiccant concentration at the outlet.

6.4 Experimental Setup

The experimental setup for studying the performance evaluation of open-cycle regenerators is shown in fig. 6.7. A solar regenerator was fabricated with a single flat galvanized iron sheet of size 1m X 0.5m and was painted with a dull black paint. This plate served as the absorber. The regenerator was inclined at 26° 19' (latitude for Dhahran) to the horizontal, facing south, so that it absorbs maximum insolation. The bottom of the absorber was insulated by 50mm thick glasswool. The supporting structure of the absorber was made of wood, while the supporting structure of the total regenerator was made of mildsteel angles. Calcium chloride solution was used as the absorbent solution and it was distributed at the top end of the regenerator through an aluminum pipe in which a slot was milled all along its length and the solution leaving the regenerator was collected at the bottom in an open channel. The absorbent solution flows over the absorber as a thin film. Thermocouples were provided in series along the width and along the length of the solar regenerator in order to record the variation in the temperature. A pump of 0.5 H.P, 37 kW, 3450

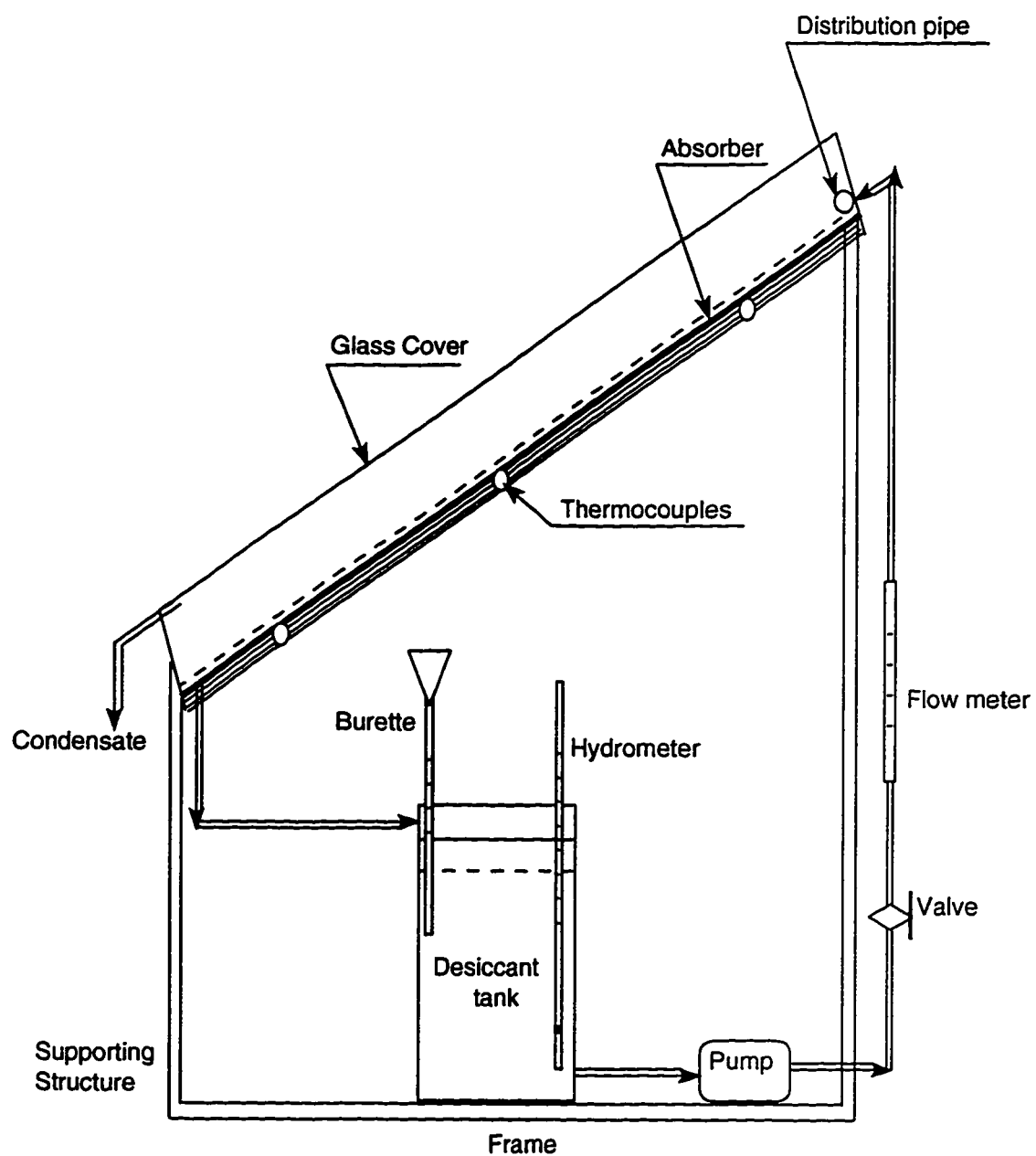


Figure 6.7: Experimental Setup for Performance Evaluation of Open-Cycle Solar Regenerators.

rpm, 60 Hz, and 220 V was used to circulate the desiccant solution from the solution tank to the collector. A flowmeter (100-1425 c³/min) was attached to the delivery side of the pump in order to record the flow rate of the desiccant over the collector surface. The setup was used for all the four types of regenerators.

In case of a fully open regenerator, as the water vapor pressure of the desiccant exceeds the water vapor pressure of the atmospheric air, mass transfer takes place from the desiccant to the atmospheric air. The solution leaving the regenerator becomes strong as a result of water evaporating from it in the regenerator. This means that the concentration of the solution increases when it is pumped back to the distribution pipe. In order to determine the rate of desorption, water was added to the circulating absorbent solution at the inlet at a controlled rate such that its concentration does not change. This was done by using a 50 ml burette by adjusting its knob and allowing the exact amount of water that was desorbed to flow into the solution tank. A hydrometer (0.700-2.000 range) was used to note the specific gravity of the desiccant solution. The hydrometer reading was also maintained to its initial reading throughout the experimental study.

The regenerator performance was studied for a fully open, fully closed, partly closed-open, or alternately closed-open regenerators depending on the type of regeneration used in the cycle. Experiments were conducted for each type of regenerator for 6 days in the month of October and November. The dates on which each experiment was conducted is shown in Table 6.1 The performance of the regenerators

Table 6.1: Dates on which Experiments were Conducted

Fully Open Regenerator	16-20 Oct.
Fully Closed Regenerator	22-26 Oct.
Partly Closed-Open Regenerator	27-31 Oct.
Alternately Closed-Open Regenerator	1-5 Nov.
Height of the Glass Cover: Fully Closed Regenerator	8-10 Nov.

was studied on the days on which the variation in insolation, ambient temperature, and the wind velocity was of not much significant.

Chapter 7

RESULTS AND DISCUSSION

Results will be discussed separately in different sections according to each topic.

7.1 Hybrid Cycle Analysis

The effect of various parameters like refrigerant (water) temperature, ambient humidity ratio, absorber temperature, and the evaporator temperature on the coefficient of performance of the hybrid cycle will be discussed. The nomograph that was developed in terms of absorber temperature, refrigerant temperature, evaporator temperature, and the regenerator temperature will also be discussed.

The effect of the ambient water (refrigerant) temperature on the coefficient of performance is shown in the Fig 7.1. Increase in the ambient water temperature tends to decrease the COP with a constant absorber and evaporator temperatures.

The increase in the ambient water temperature has the effect of reduction in the rate of vaporization in the evaporator which results in decrease in the capacity to absorb more heat from the cold storage or the space being cooled. It is also obvious that the most favorable range of variation is in the region of reduced ambient water temperature. A maximum COP of 1.204 is obtained for the absorber temperature of 40°C , evaporator temperature of 22°C , and refrigerant temperature of 30°C . As the regenerator temperature increases, the COP also increases. However, the increase in the regenerator temperatures is insignificant beyond certain COP values.

Figure 7.2 shows the variation of COP with the absorber temperature and the effect of the ambient humidity. As the ambient air becomes more humid the desiccant has to absorb more moisture, which tends to increase in the latent load. This results in dilution of the desiccant which requires higher regeneration temperatures, finally affecting the overall COP of the system. The COP of the hybrid system decreases with an increase in the humidity ratio. For a typical humidity ratio of 0.020 the maximum COP obtained was 1.253 for an absorber temperature of 35°C , ambient temperature of 30°C , and the evaporator temperature of 22°C . As the absorber temperature increases, the COP decreases.

From Fig. 7.3, it can be seen that as the absorber temperature increases the optimum regenerator temperature also increases. For an absorber temperature of 35°C , a desiccant optimum regeneration temperature is 77°C whereas 79°C is the optimum temperature for an absorber temperature of 40°C . This increase in the temperature

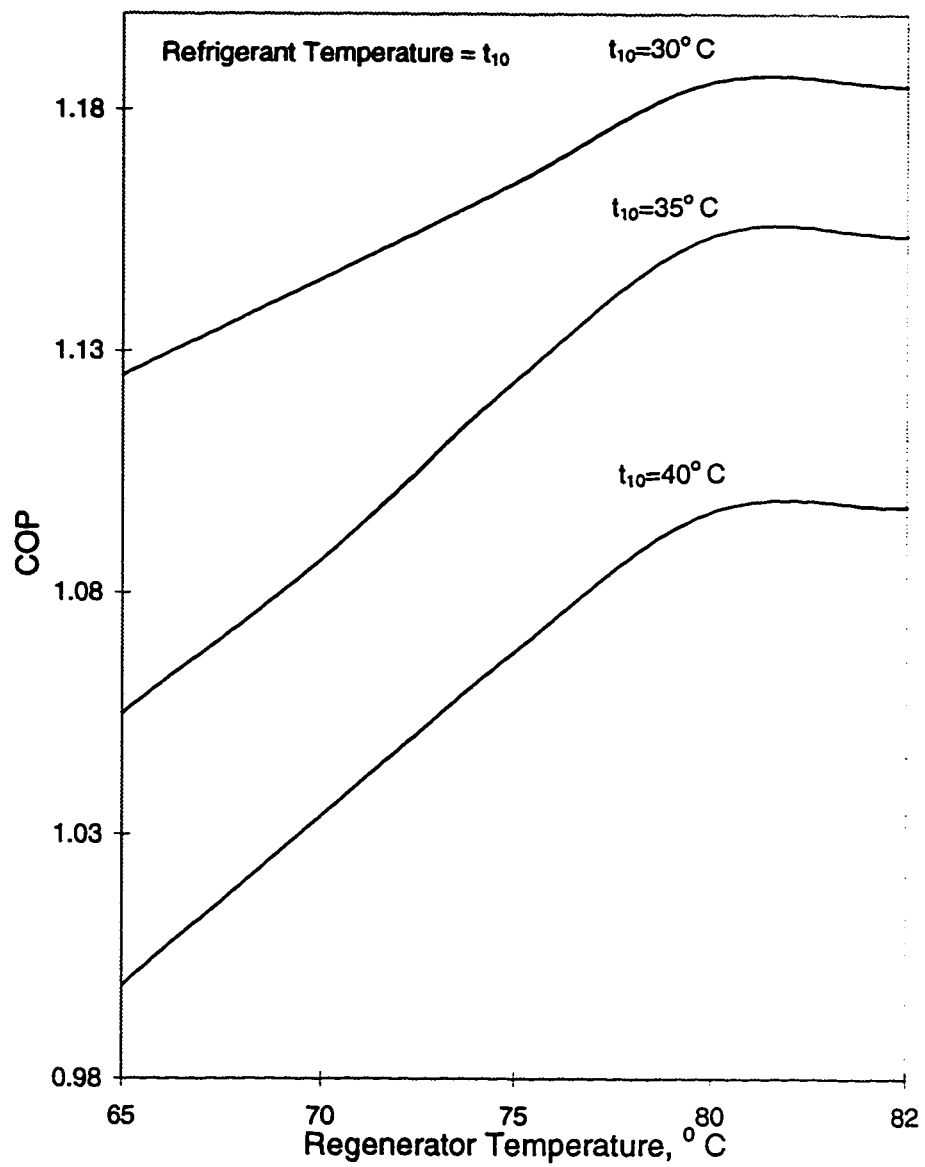


Figure 7.1: Variation of COP with Regenerator Temperature for Different Refrigerant Temperatures.

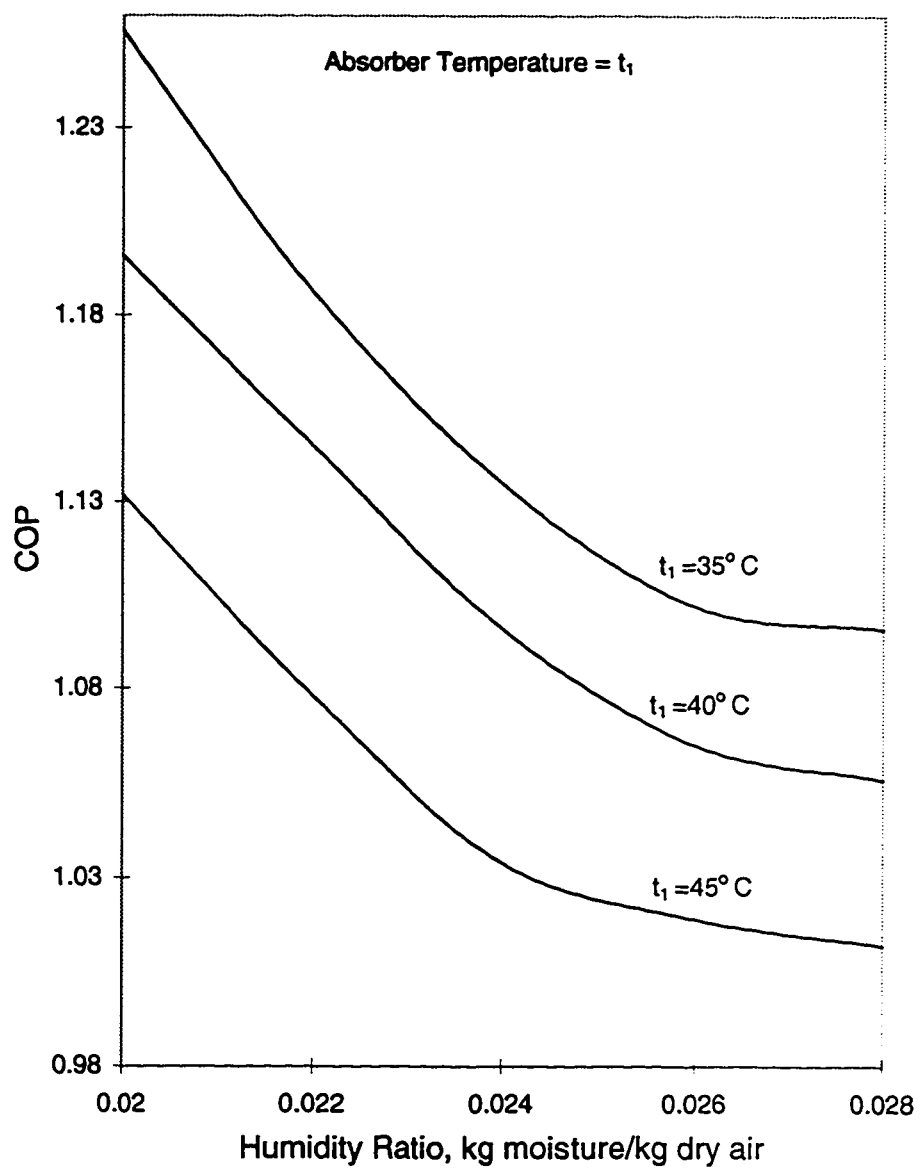


Figure 7.2: Variation of COP with Humidity Ratio for Different Absorber Temperatures.

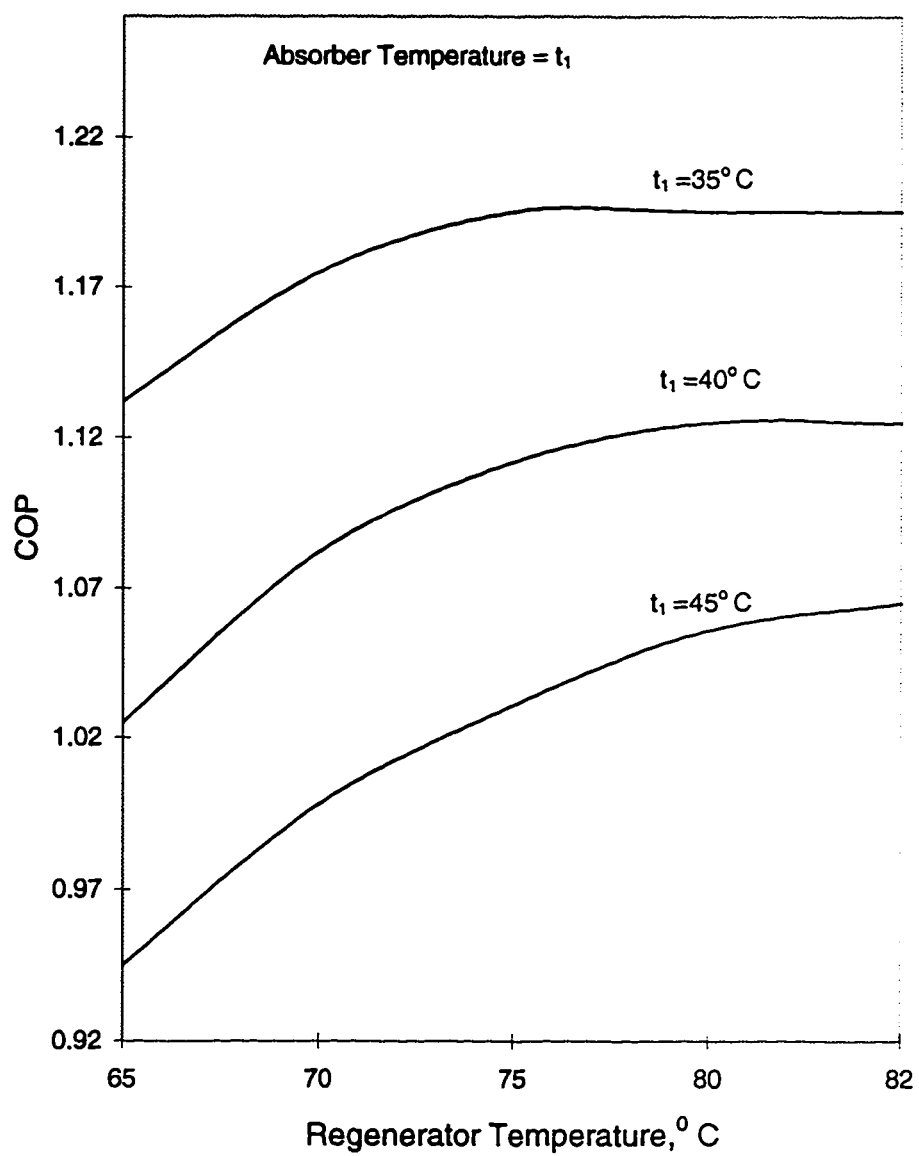


Figure 7.3: Variation of COP with Regenerator Temperature for Different Absorber Temperatures

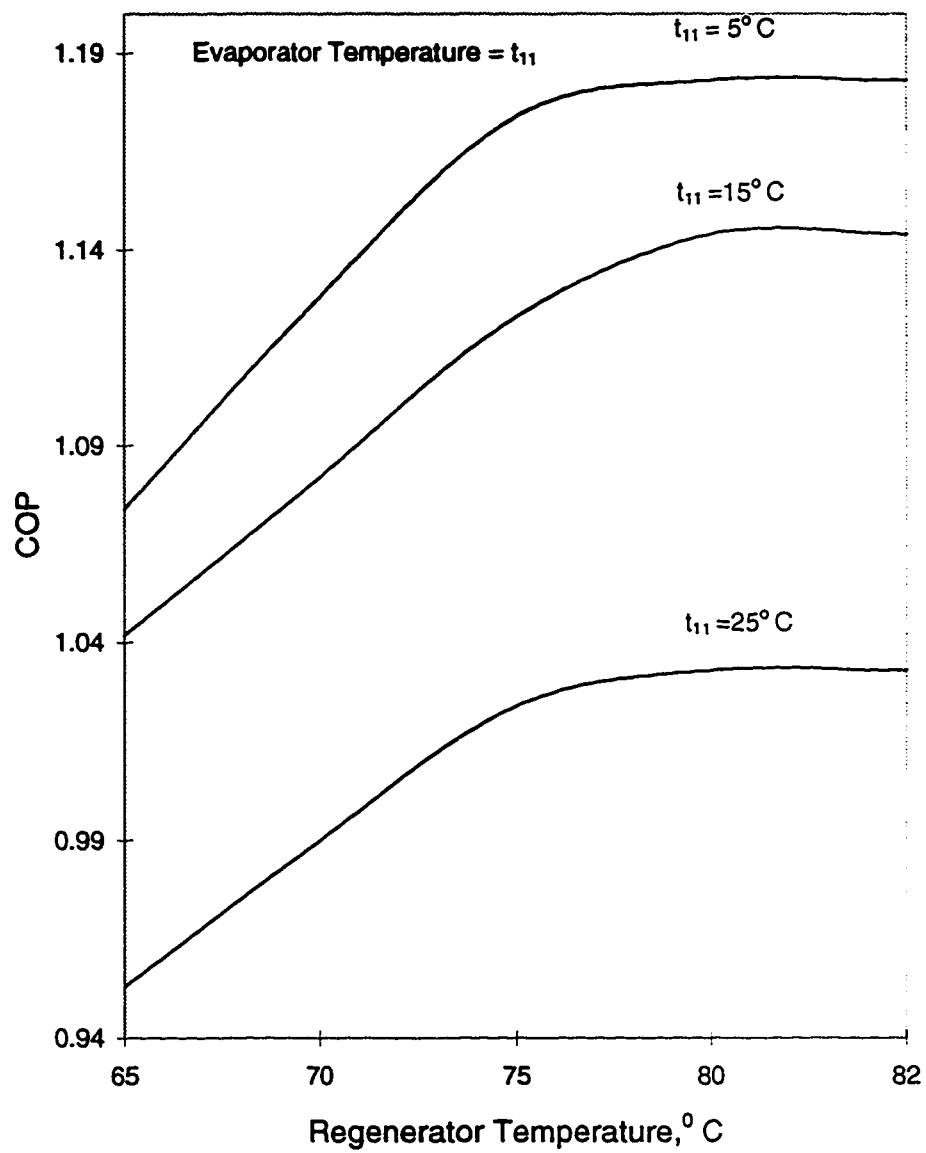


Figure 7.4: Variation of COP with Regenerator Temperature for Different Evaporator Temperatures.

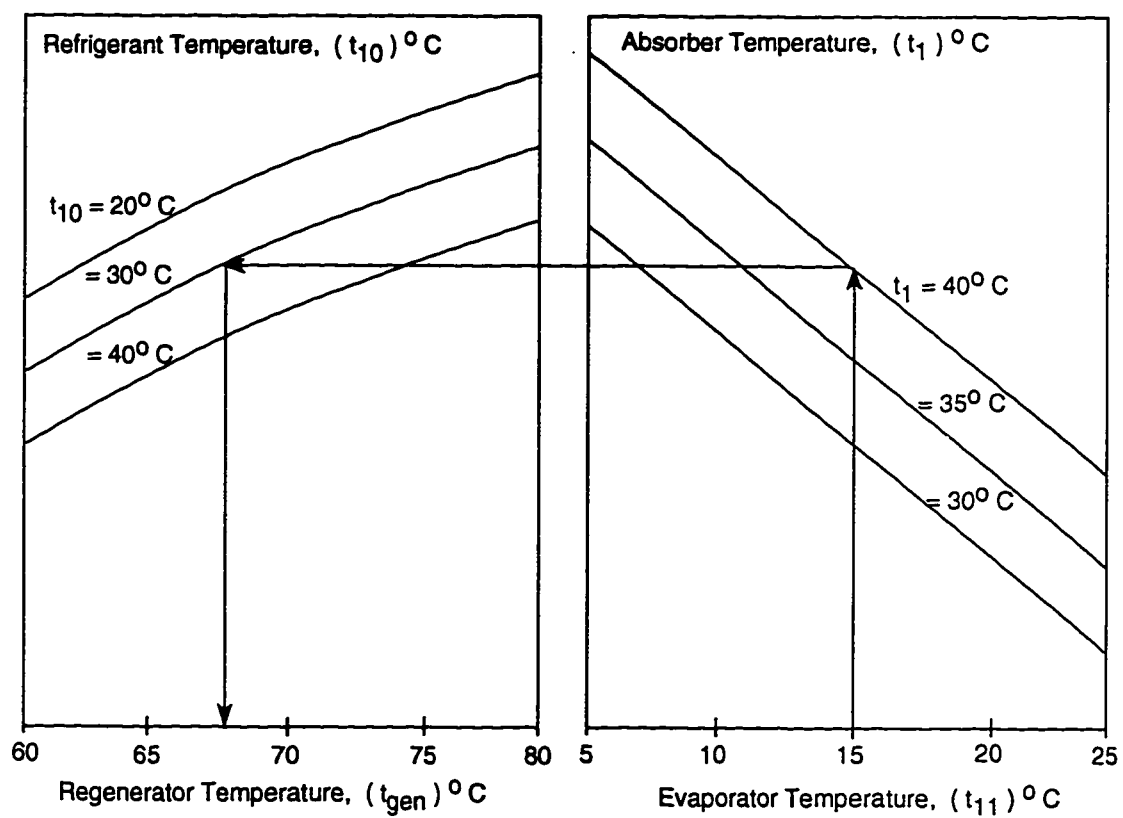


Figure 7.5: Nomograph for Determining the Minimum Temperature Required in the Regenerator

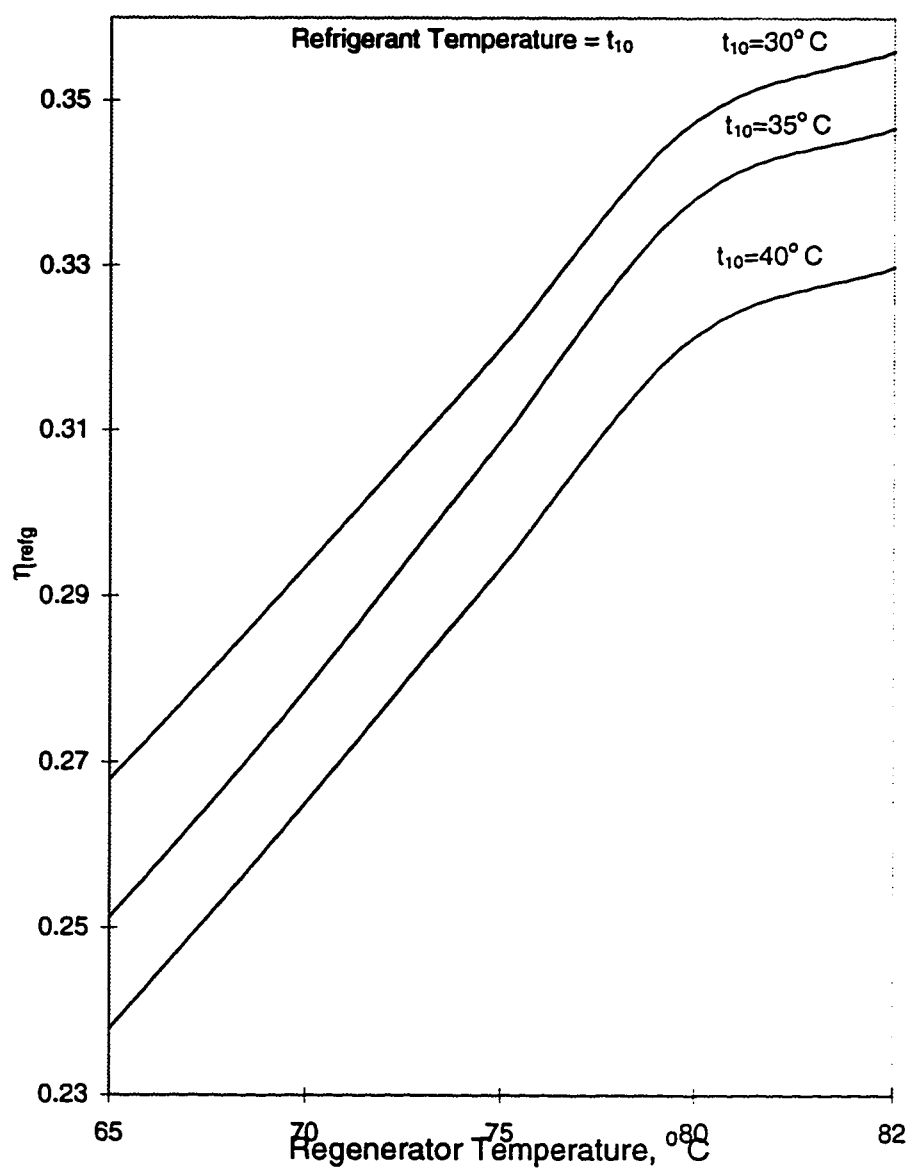


Figure 7.6: Variation of Refrigerating Efficiency with Regenerator Temperature for Different Refrigerant Temperatures .

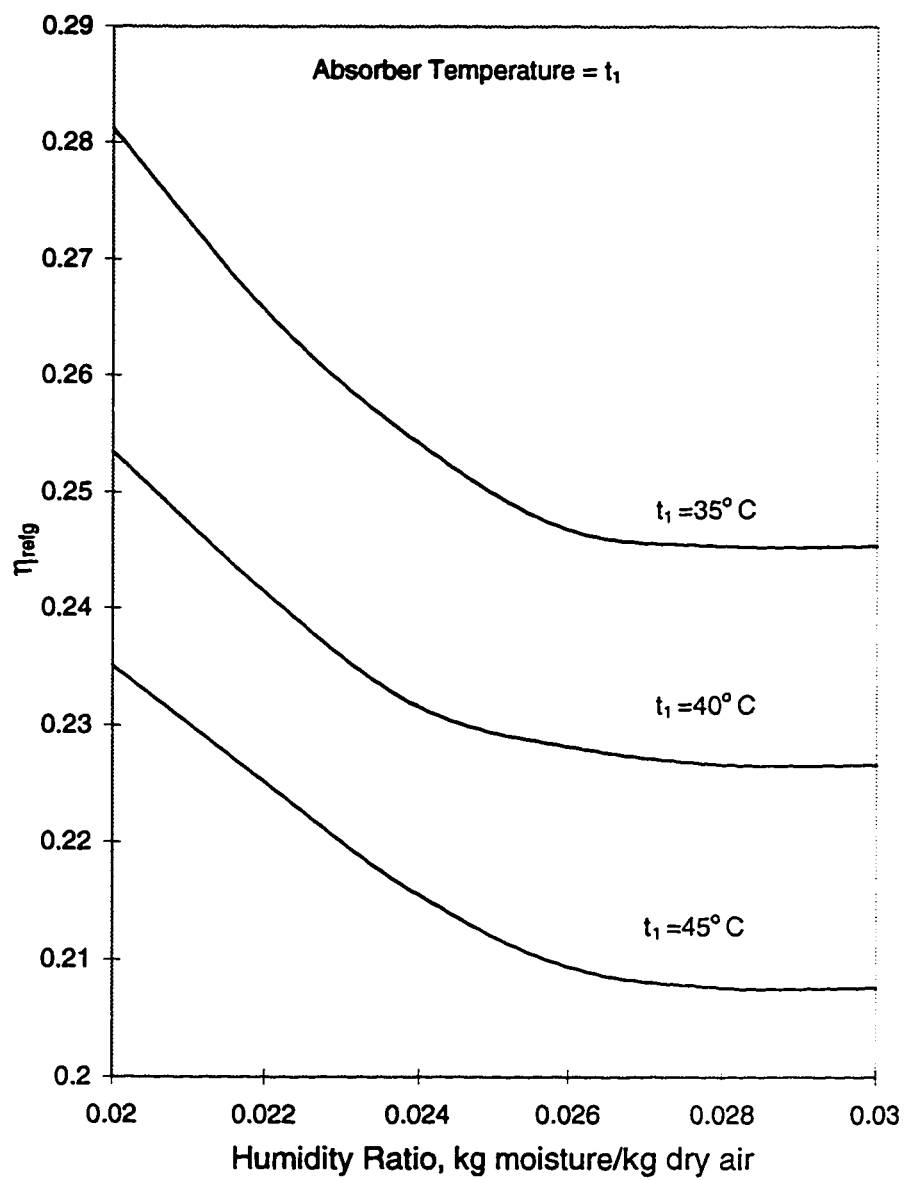


Figure 7.7: Variation of Refrigerating Efficiency with Humidity Ratio for Different Absorber Temperatures.

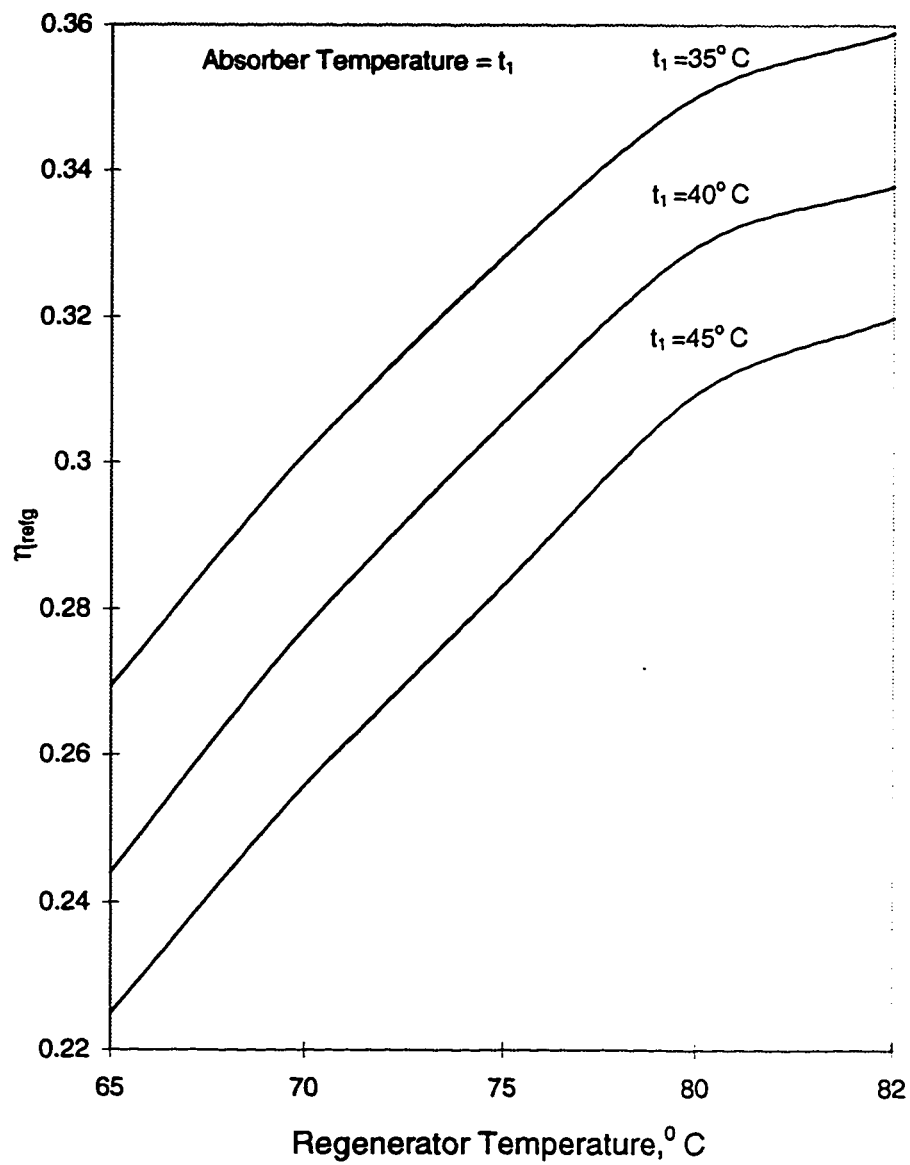


Figure 7.8: Variation of Refrigerating Efficiency with Regenerator Temperature for Different Absorber Temperatures.

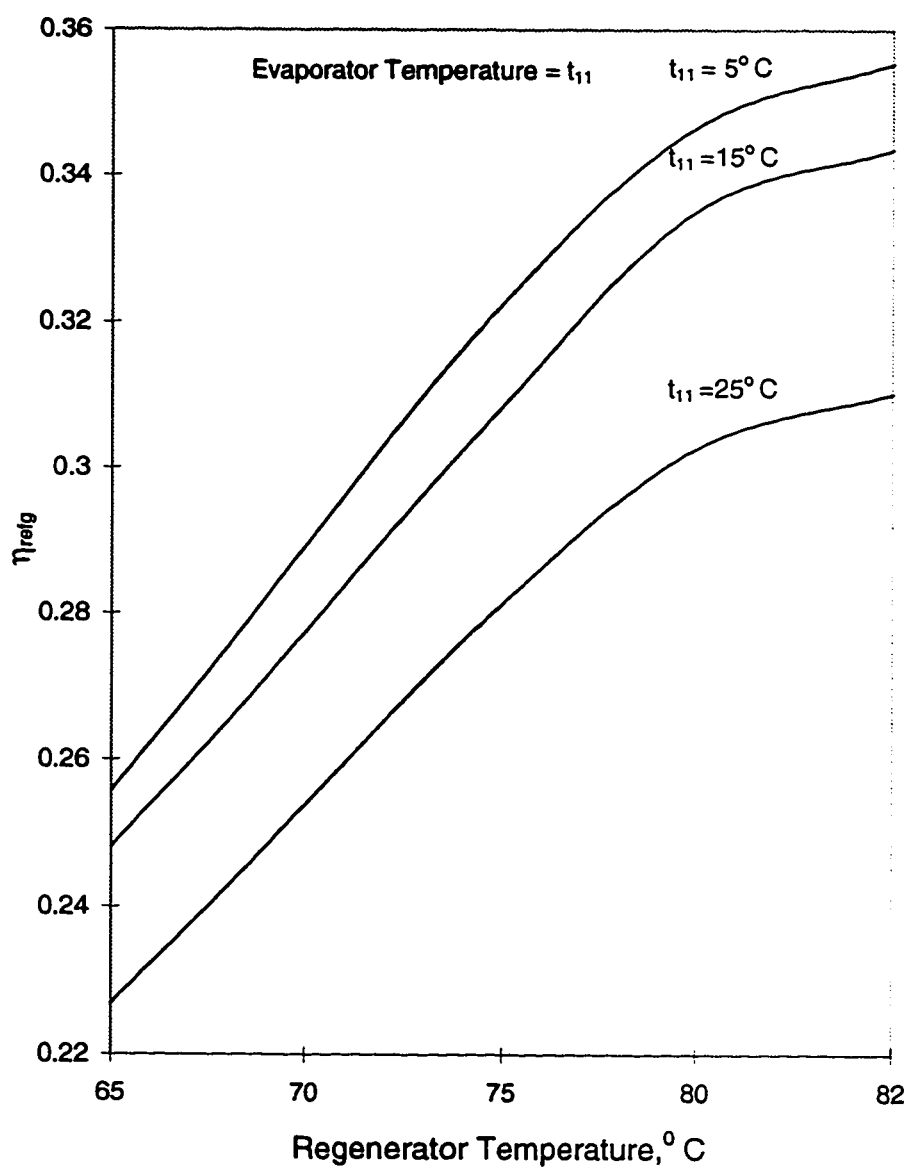


Figure 7.9: Variation of Refrigerating Efficiency with Regenerator Temperature for Different Evaporator Temperatures .

of regeneration is due to increase in the rate of absorption in the absorber which results in high temperature in the absorber. Hence the temperature required to regenerate this increase in the absorbed vapor in the regenerator is higher. The maximum COP obtained for this variation was 1.20 with an optimum temperature of 77°C required for regeneration of the weak desiccant at an absorber temperature of 35°C .

With an increase in the evaporator and the regeneration temperatures the coefficient of performance of the hybrid unit is found to increase with constant absorber and ambient water (refrigerant) temperatures. Also, when the boiling point of the water in the evaporator decreases, the regenerator temperature rises with constant ambient water temperature and absorber temperature. This is explained in Fig. 7.4 by the fact that with an increase in evaporation in the evaporator, the concentration of the solution in the absorber decreases, as a result of which the regenerator temperature increases. Hence a higher evaporation in the evaporator will require higher temperatures for regeneration of the weak desiccant solution. Therefore as the evaporator temperature decreases the COP of the hybrid cycle also decreases. A nomograph is developed to determine the minimum regenerator temperature required for the system operation. Figure 7.5 shows that for $T_{11} = 15^{\circ}\text{C}$, $T_1 = 40^{\circ}\text{C}$, and $T_{10} = 30^{\circ}\text{C}$, the minimum temperature required for regeneration is around 68°C .

The coefficient of performance of the hybrid cycle has been modified by compar-

ing it with that of the carnot cycle coefficient of performance and the results are presented in Figs. 7.6, 7.7, 7.8, and 7.9.

From the above discussion, it is obvious that the COP of the hybrid cycle due to the variations in the different parameters range between 0.96 and 1.25. Hence this hybrid cycle has the tendency to take into considerations the different effects of ambient temperature and humidity without much sacrifice in the overall performance of the system. Thus it is the most suitable hybrid machine in regions which are mostly prone to sudden variations in high temperatures and humidities.

7.2 Exergy Analysis

The variation of irreversibilities with respect to the variation of different parameters like insolation, desiccant inlet temperature, desiccant inlet concentration, ambient vapor pressure, ambient wind velocity, ambient temperature, and desiccant mass flow rate will be discussed in this section.

Table 4.1 in chapter 3, gives the values of the parameters at different state points of the hybrid cycle. The parameters of the partly closed-open solar regenerator that have been maintained constant for variation of different parameters are shown in Table 7.1. Each of these parameters are varied and its consequent effect on the irreversibilities are studied, keeping the rest of the other parameters constant.

Table 7.1: Regenerator Parameters

Insolation, I	3000 kJ/h m ²
Ambient Temperature, t_0	35 °C
Desiccant Inlet Temperature, t_i	30 °C
Desiccant Inlet Concentration, ξ	40%
Ambient Vapor Pressure, P_0	20 mm Hg
Ambient Wind Velocity, V_e	4.5 m/s

Figure 7.10 shows the variation of the irreversibilities with the desiccant flow rate and the ambient temperature for $I = 3000 \text{ kJ/h m}^2$, $t_i = 30^\circ\text{C}$, $P_0 = 20 \text{ mm Hg}$, $\xi = 40\%$, and $V_e = 4.5 \text{ m/s}$. The irreversibilities are found to approach a maximum value as the flow rate approaches zero or infinity. When the flow rate is zero there is no interaction with the desiccant and ambient, therefore the entire insolation is lost to the ambient. The irreversibilities in this case reaches its peak value. Likewise, when the flow rate tends to infinity the collector temperature is almost declined to the desiccant inlet temperature and the irreversibility rate is again maximum since in this case no collector exits at such low temperatures. For the desiccant mass flow rate about 30 kg/h m^2 and for the ambient temperatures considerably greater than 40°C , the irreversibilities are substantially higher than what it would be if no collector were present. In otherwords, the area above each of the irreversibility curves represents the potential for exergy extraction from the collector. This demonstrates graphically that minimizing irreversibilities is equivalent to maximizing the exergy output from the desiccant-regenerator system. The effect of insolation on the irreversibility rate is shown in Fig. 7.11. The irreversibilities are found to increase with the increase in the insolation for $t_0 = 35^\circ\text{C}$, $t_i = 30^\circ\text{C}$, $P_0 = 20 \text{ mm Hg}$, $\xi = 40\%$, and $V_e = 4.5 \text{ m/s}$. This is due to the fact that, there is a high interaction between the ambient, insolation, and the desiccant solution. As the flow rate is reduced, the interaction between the insolation and the desiccant also increases but the interaction between the insolation and ambient is at a higher rate. Therefore most of the insolation and

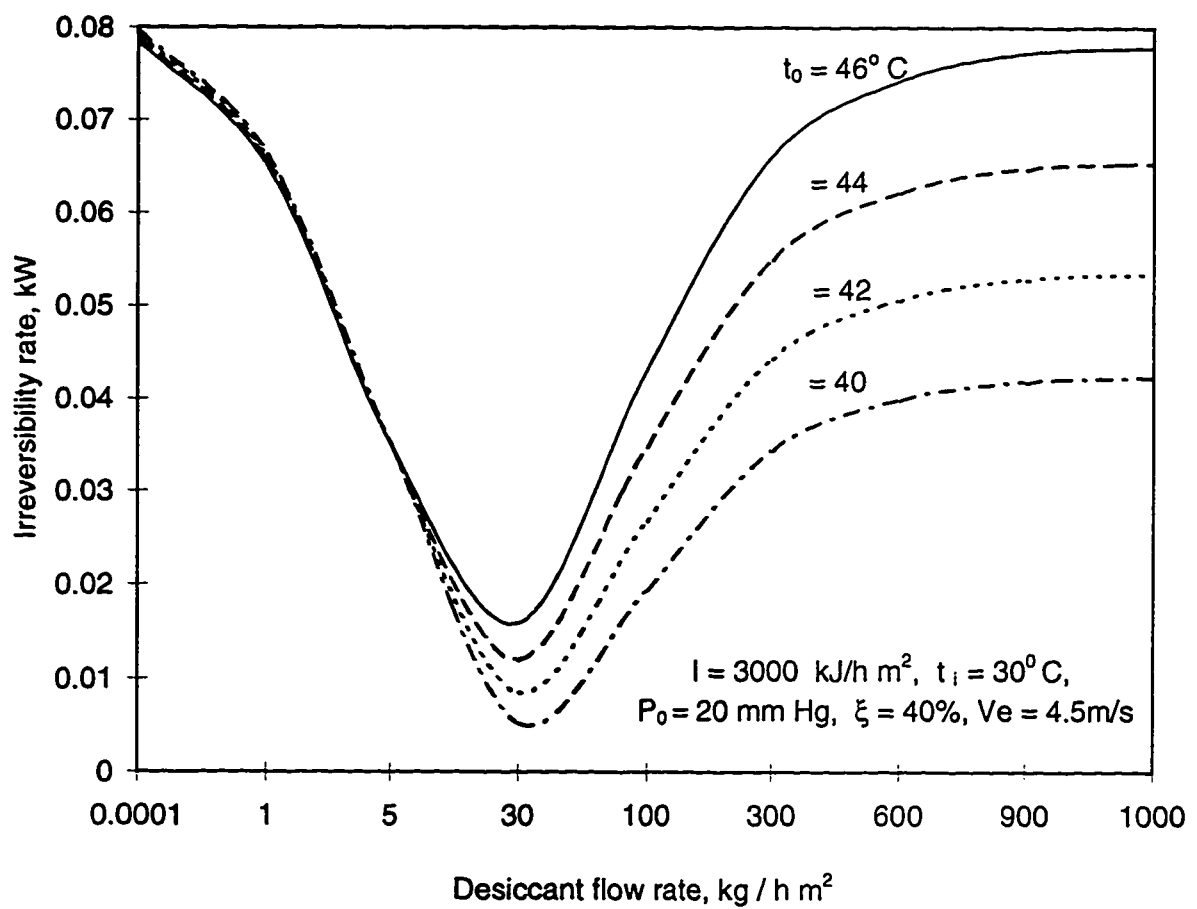


Figure 7.10: Effect of Desiccant Mass Flow Rate on Irreversibilities as a Function of Ambient Temperature.

the desiccant solution exergy is lost to the ambient, the rate of irreversibilities being further increased. The rate of irreversibilities are found to be more pronounced at low mass flow rates of the desiccant solution. Figure. 7.12 shows the variation of irreversibilities with the desiccant inlet concentration and mass flow rate. It is found that increase in the inlet concentration of desiccant solution decreases the rate of irreversibilities for $t_0 = 35^\circ\text{C}$, $t_i = 30^\circ\text{C}$, $P_0 = 20 \text{ mm Hg}$, $I = 3000 \text{ kJ/h m}^2$, and $V_e = 4.5 \text{ m/s}$. It can be inferred that as the inlet concentration of the desiccant solution increases its ability to absorb more insolation decreases, that is its interaction with the ambient decreases. This has an effect of exergy retention by the regenerator which in turn has an effect of decrease in the irreversibility rate of the regenerator. At low flow rates, the interaction between the desiccant and the ambient is found to increase, the irreversibility rate is thus higher for low flow rates than when compared to that at higher desiccant mass flow rates.

With an increase in the inlet solution temperature, its interaction with the ambient also increases for $t_0 = 35^\circ\text{C}$, $\xi = 40\%$, $P_0 = 20 \text{ mm Hg}$, $I = 3000 \text{ kJ/h m}^2$, and $V_e = 4.5 \text{ m/s}$. This is explained in Fig. 7.13 by the fact that most of the exergy is lost to the ambient because of the solution's high temperatures. Also, as the solution temperature increases, its capacity to release the moisture increases. Therefore there is an overall effect of the exergy reduction due to increase in its temperature as well as decrease in the rate of desiccant flow rate. This tends to increase the rate of irreversibilities. At lower flow rates and high inlet temperatures, the exergy lost

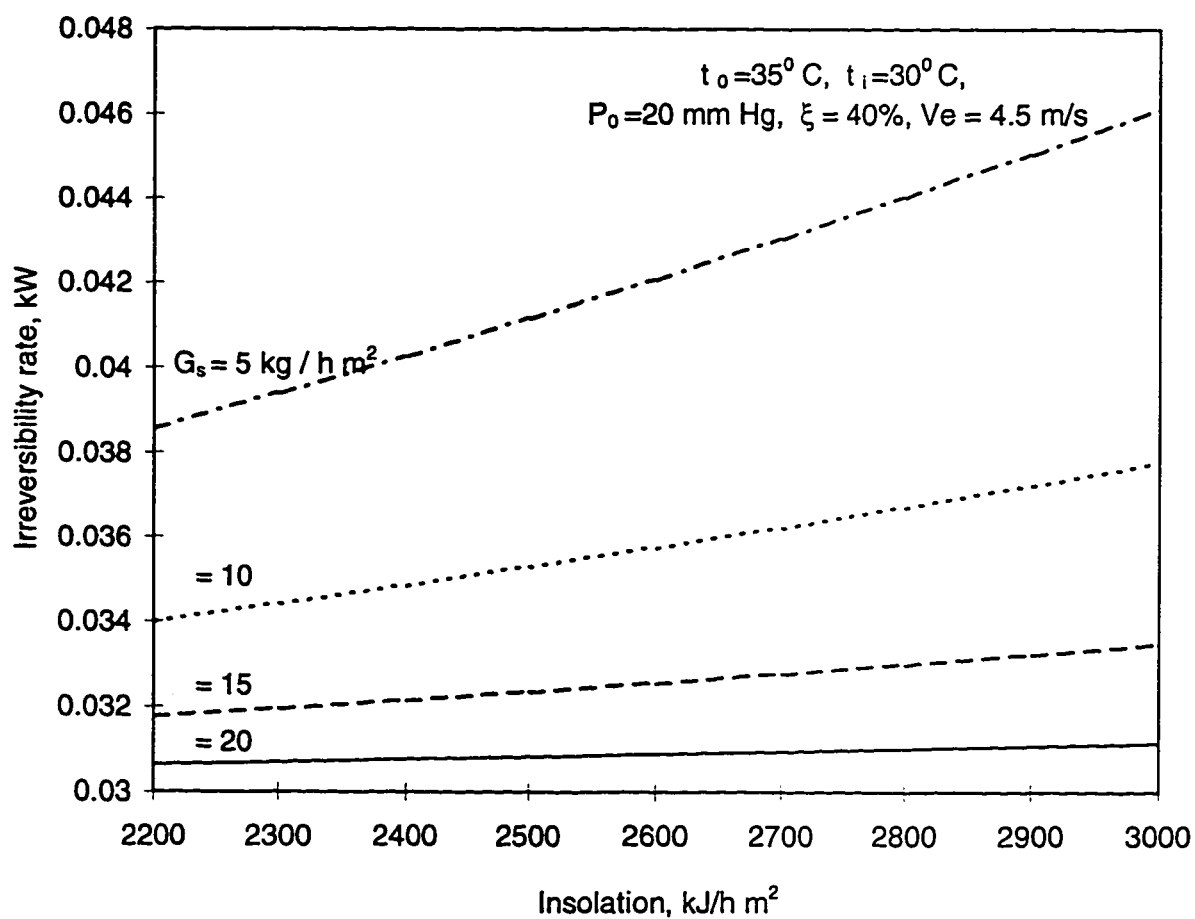


Figure 7.11: Effect of Insolation on Irreversibilities as a Function of Mass Flow Rate.

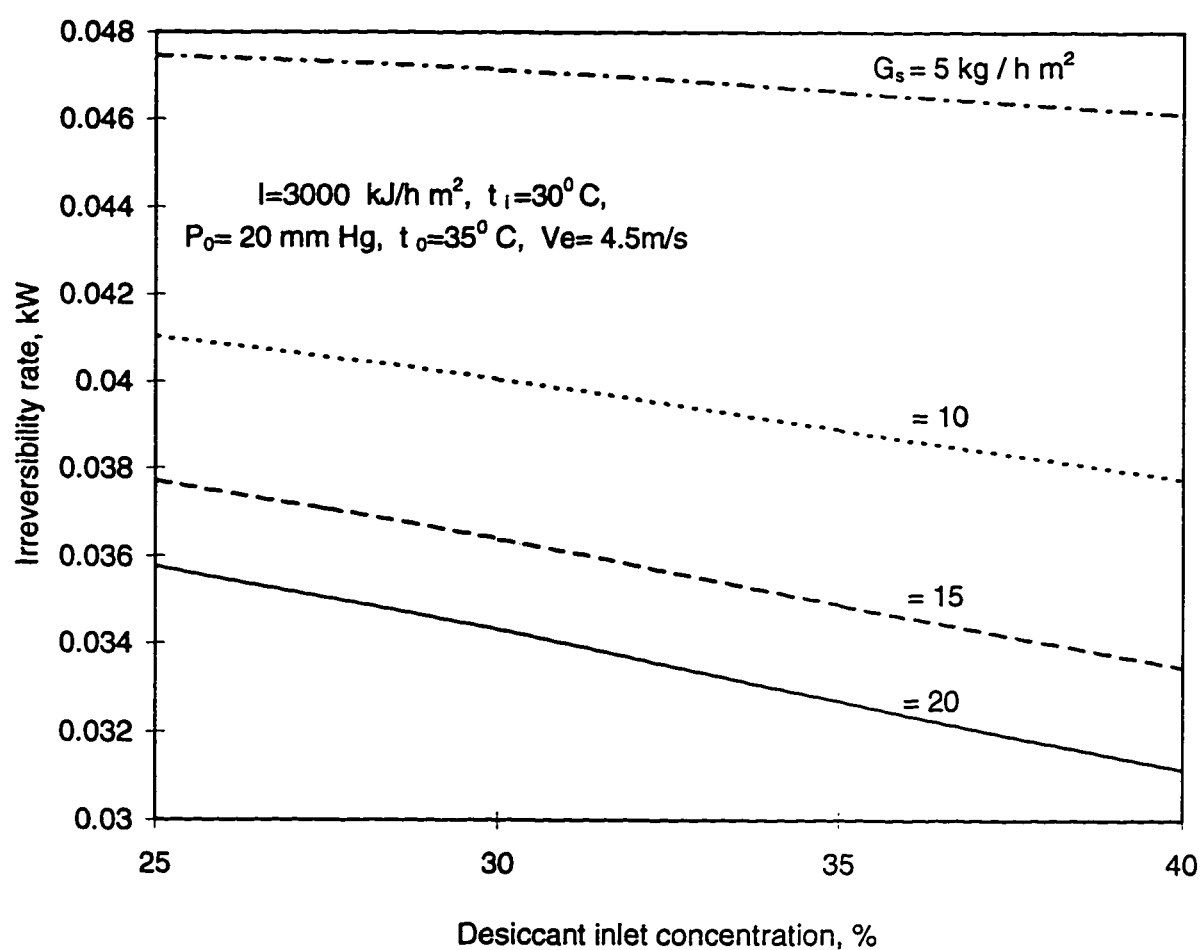


Figure 7.12: Effect of Desiccant Inlet Concentration on Irreversibilities as a Function of Mass Flow Rate.

is maximum which in turn have an effect of increase in irrerversibilities.

The effect of ambient vapor pressure and the irreversibility rate is shown in the Fig. 7.14 for $t_0 = 35^\circ\text{C}$, $\xi = 40\%$, $t_i = 30^\circ\text{C}$, $I = 3000\text{kJ/h m}^2$, and $V_e = 4.5\text{m/s}$. As the ambient vapor pressure increases, the desiccant's ability to loose moisture to the atmosphere decreases because of the decrease in the vapor pressure difference. Thus, the rate of moisture evaporation from the regenerator will decrease. This phenomena will have an effect of reduction in the output exergy of the collector which implies that the rate of irreversibility from the regenerator increases.

The effect of ambient temperature on irreversibilities is shown in Fig. 7.15 for $P_0=20\text{ mm Hg}$, $\xi=40\%$, $t_i=30^\circ\text{C}$, $I=3000\text{kJ/h m}^2$, and $V_e=4.5\text{m/s}$. At low solution flow rates the interaction with insolation, ambient temperature and the desiccant solution is high. As the mass flow rates increases the desiccant solution will have an advantage of having higher interactions with the increase in the ambient temperatures, resulting in the increase of irreversibilities. For flow rates as low as 5kg/h m^2 , the solution is found to have less interaction with the ambient, which results in almost constant irreversibility rate. Figure 7.16 shows that the wind velocity increases the irreversibility rate in the regenerator for $P_0 = 20\text{ mm Hg}$, $\xi = 40\%$, $t_i = 30^\circ\text{C}$, $I = 3000\text{kJ/h m}^2$, and $t_0 = 35^\circ\text{C}$. This can be expressed by the fact that an increase in wind velocity has an effect of higher interactions with the desiccant solution and the ambient. Hence most of the insolation as well as the collector heat is carried away by the convection. The figure shows that the wind velocity affects

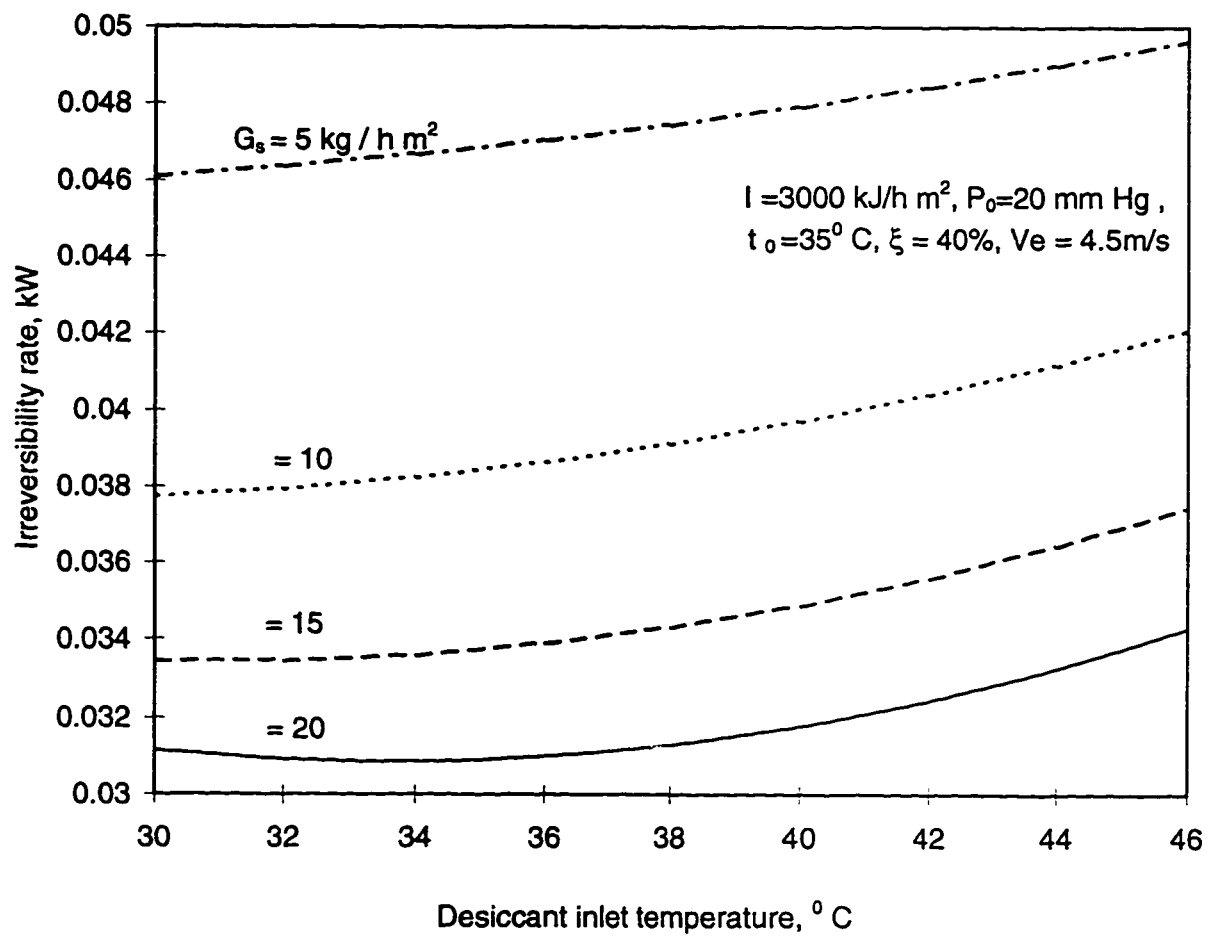


Figure 7.13: Effect of Desiccant Inlet Temperature on Irreversibilities as a Function of Mass Flow Rate.

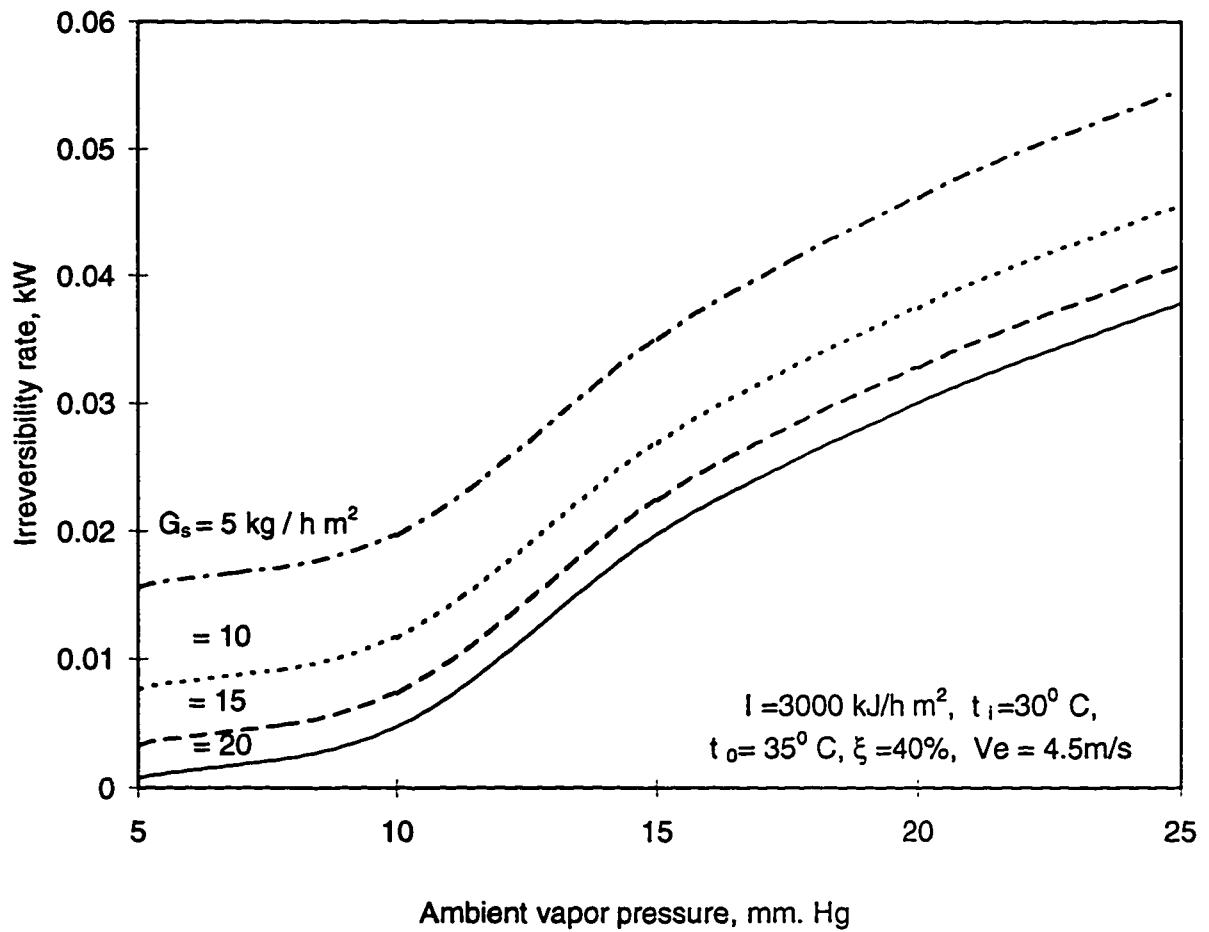


Figure 7.14: Effect of Ambient Vapor Pressure on Irreversibilities as a Function of Mass Flow Rate.

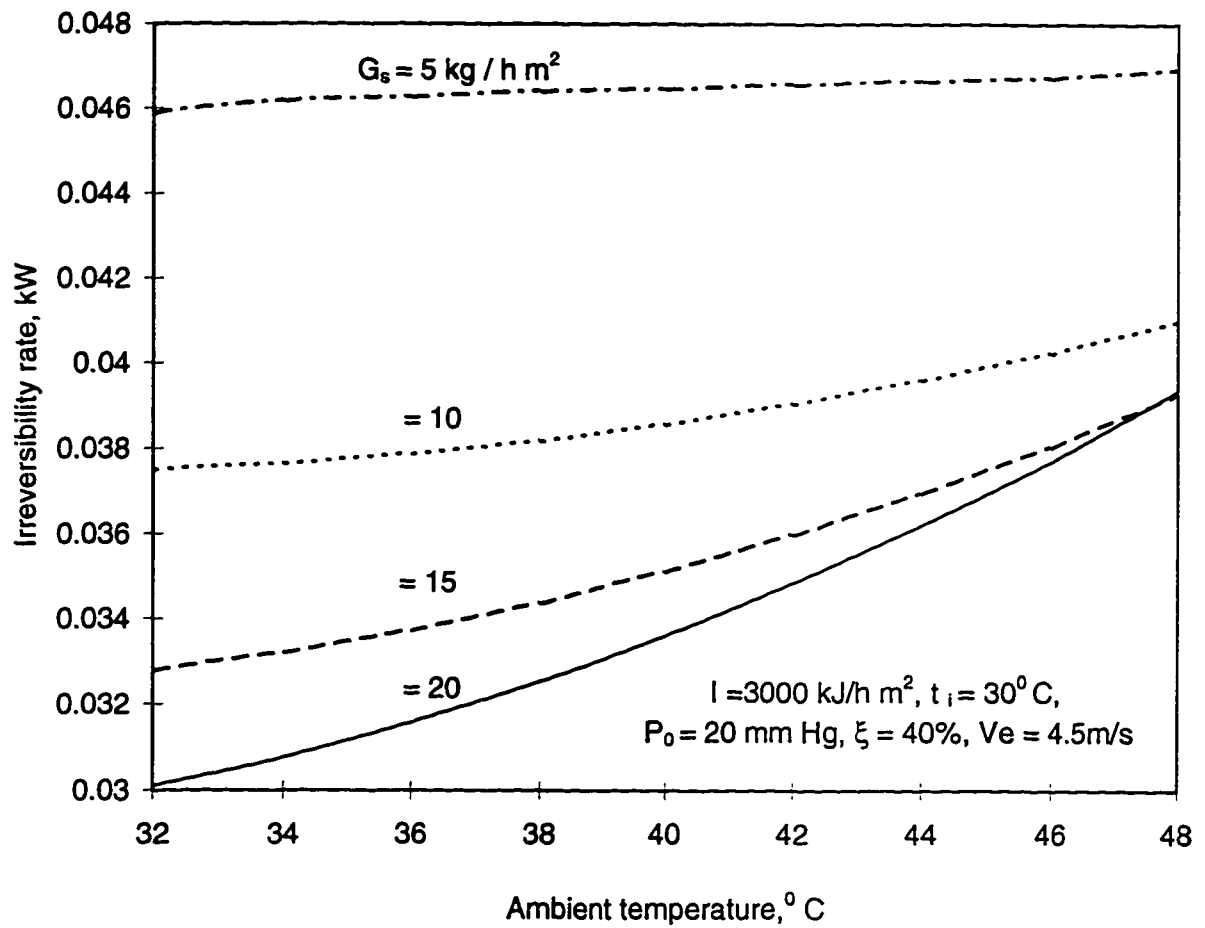


Figure 7.15: Effect of Ambient Temperature on Irreversibilities as a Function of Mass Flow Rate.

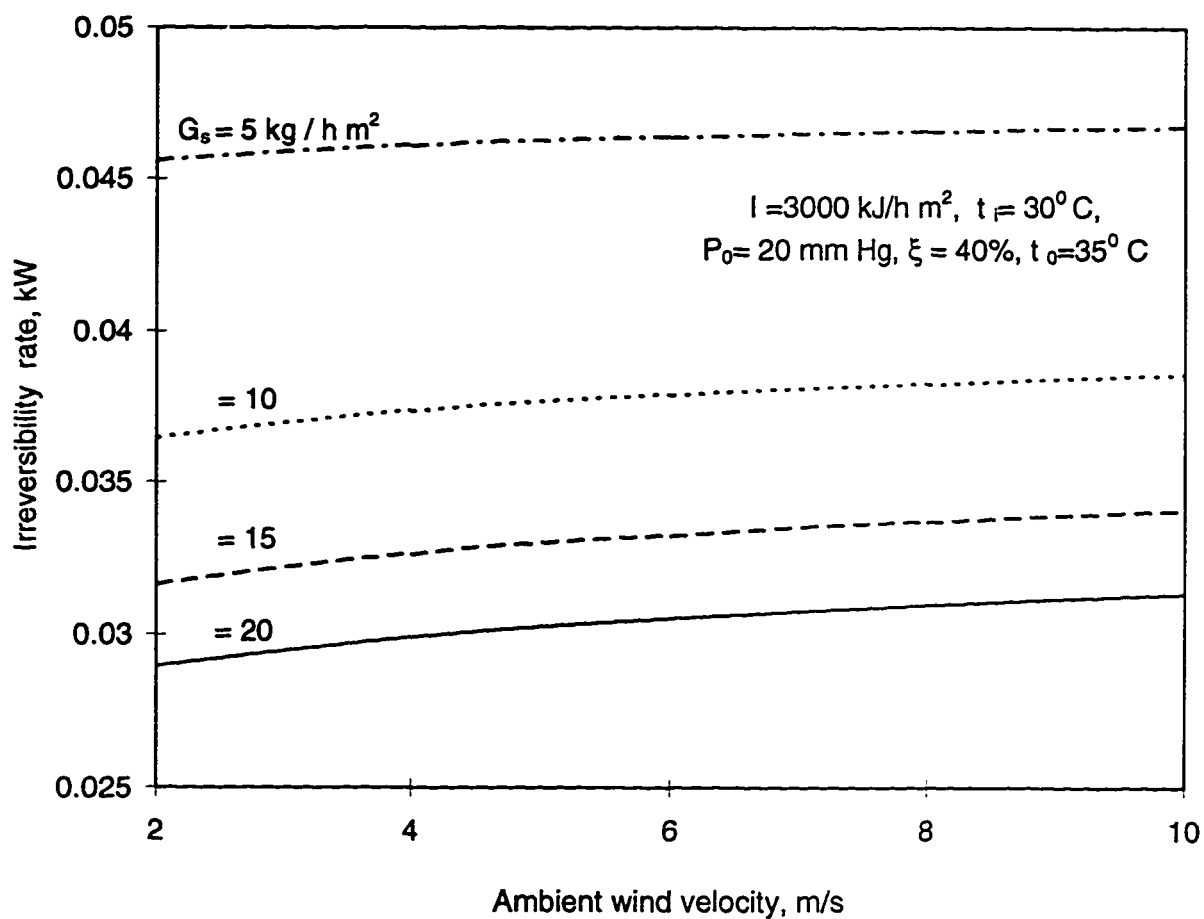


Figure 7.16: Effect of Ambient Wind Velocity on Irreversibilities as a Function of Mass Flow Rate.

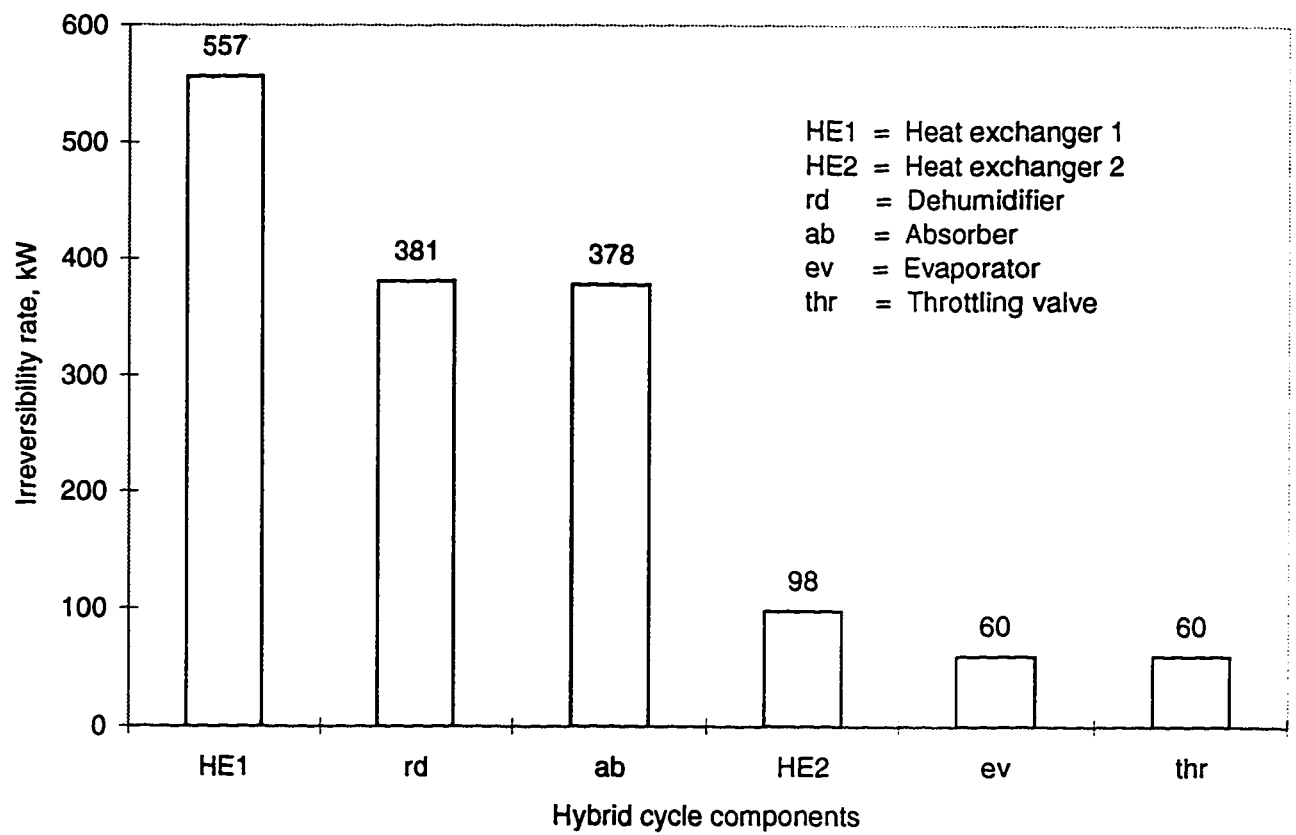


Figure 7.17: Irreversible Losses in the Hybrid Cycle Components.

more at low desiccant mass flow rates thereby enhancing further the irreversibility rate of the regenerator.

The results of the irreversible losses for evaporator, absorber, heat exchanger 1, heat exchanger 2, throttling valve, and the dehumidifier are shown in Fig. 7.17. The irreversibilities are found to be almost similar in the absorber and the dehumidifier. This is due to the reason that both these components have the function of absorption of moisture. The irreversibilities in heat exchanger 1 are more than heat exchanger 2, because heat exchanger 1 is operating at higher temperatures.

7.3 Experimental Analysis of Solar Regenerators

The results of the experiment results that were conducted on four different types of regenerators will be discussed in this section. The four different types of regenerators that were studied were fully open, fully closed, partly closed-open, and alternately closed-open. Each experiment for individual regenerator was conducted for a 6 day period, on an hourly basis and the results that were closest for two different days were studied, since the variation of parameters like insolation, ambient temperature, ambient wind velocity was found to have negligible effect for these two days. The variation of moisture evaporation rate on an hourly basis on different days for different regenerators and the temperature profiles for these four different

types of regenerators along the length of the regenerator, will also be discussed. The variation of moisture evaporation rate as a function of glass cover height from the regenerator plate was also studied.

Figure 7.18 shows the variation of temperature along the plate length of a fully open regenerator on Oct. 18 and 19, 1995. The regenerator was fully open to atmosphere and had direct interactions with the ambient. The rate of moisture evaporation is shown in Fig. 7.19. The rate of moisture evaporation in this type of regenerator was found to be maximum when compared to the fully closed, partly closed-open, and the alternately closed-open regenerators since the desiccant solution surface was fully open to the convecting ambient. The moisture was evaporated and carried away by the convecting ambient as soon as it was formed. The maximum moisture evaporation rate was 0.159 kg/h m^2 at a regenerator plate temperature of about 74°C . The variation of moisture evaporation along the time of the day is shown in Table 7.2.

The variation of the temperature along a fully closed regenerator on Oct. 25 and 26 is shown in Fig. 7.20. The regenerator was fully closed by a glass cover and had no direct interactions with the ambient. The moisture evaporated from the regenerator condensed under the glass cover plate and was collected in a separate container. At lower heights of the glass cover plate from the absorbing surface, the condensate was minimum. As the height of the glass cover was raised, the condensate also increased. The height of glass cover plate from the absorbing surface was varied from 0.35m

to 0.65m. It was found that the condensate was minimum at 0.35m height and was found to maximum at 0.45m from the absorbing surface. The condensate was found to be almost similar for any further increase in the glass cover plate. The variation of moisture evaporation with time is shown in Fig.7.21, while Fig.7.22 shows the variation of moisture evaporation rate with glass cover height from the absorbing surface on Nov. 8, 9, 10, 1995. A similar analysis for the moisture evaporation rate with the change in glass cover height was performed on June 6, 7, 8, and 9 1996 and the results are presented in Fig.7.23. The experiment was reconducted to get better results for uniform heights of the glass cover from the absorber plate. It is found that the condensation in this case started at a glass cover height of 0.4m. This is due to reason that the insolation and the ambient temperatures in the month of June 1996 were found to be higher than that in the month of Nov. 1995. Therefore the glass cover plate increased and the condensation at lower glass cover heights decreased. A maximum regenerator plate temperature of about 88°C was found in this type of regenerator. The variation of moisture evaporation with the time of the day is shown in Table 7.3. The rate of moisture evaporation was found to be minimum inspite of desiccant solution temperatures when compared to the other regenerators since the desiccant solution surface was fully closed and there was no convecting ambient.

The effect of variation of time on moisture evaporation rate for a partly closed-open solar regenerator on Oct. 29 and 30 is shown in Fig. 7.24. Part of this

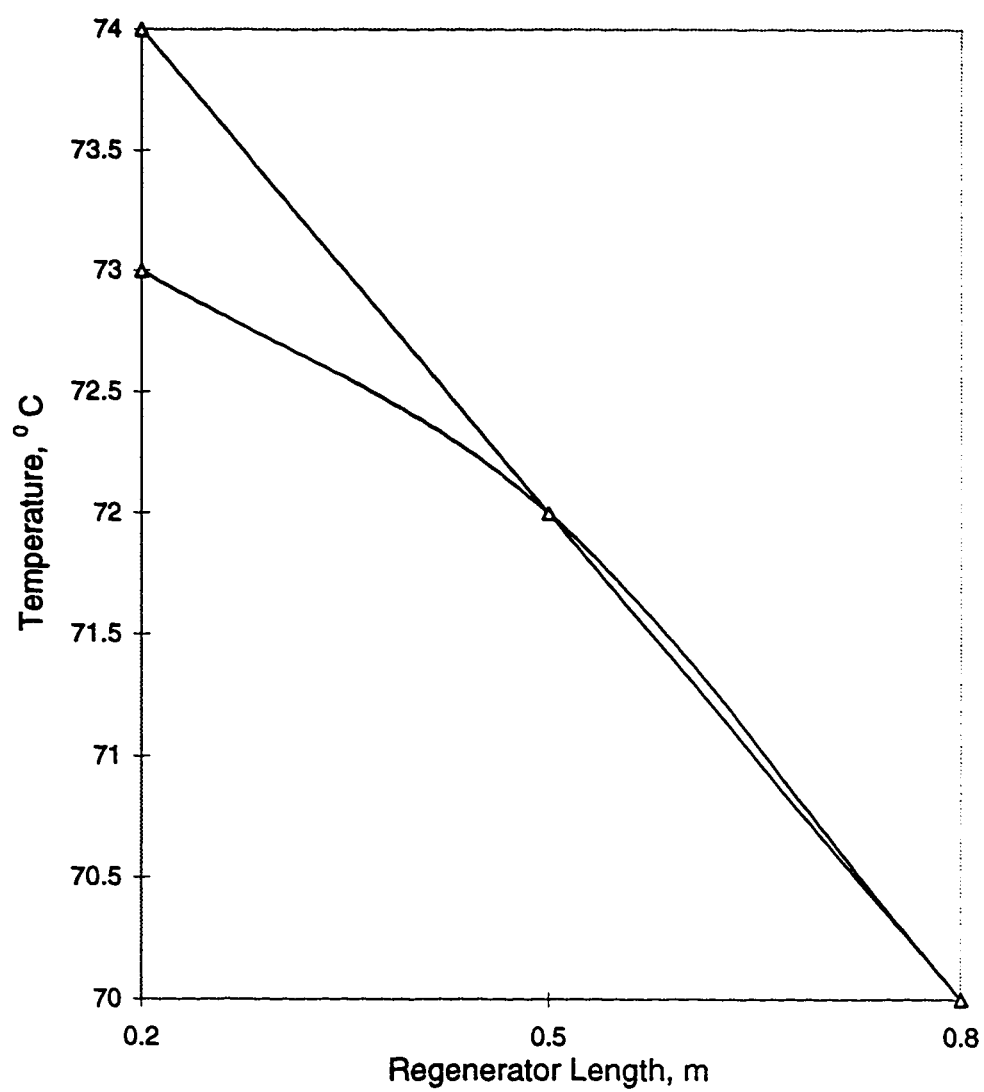


Figure 7.18: Temperature Profile in a Fully Open Solar Regenerator.

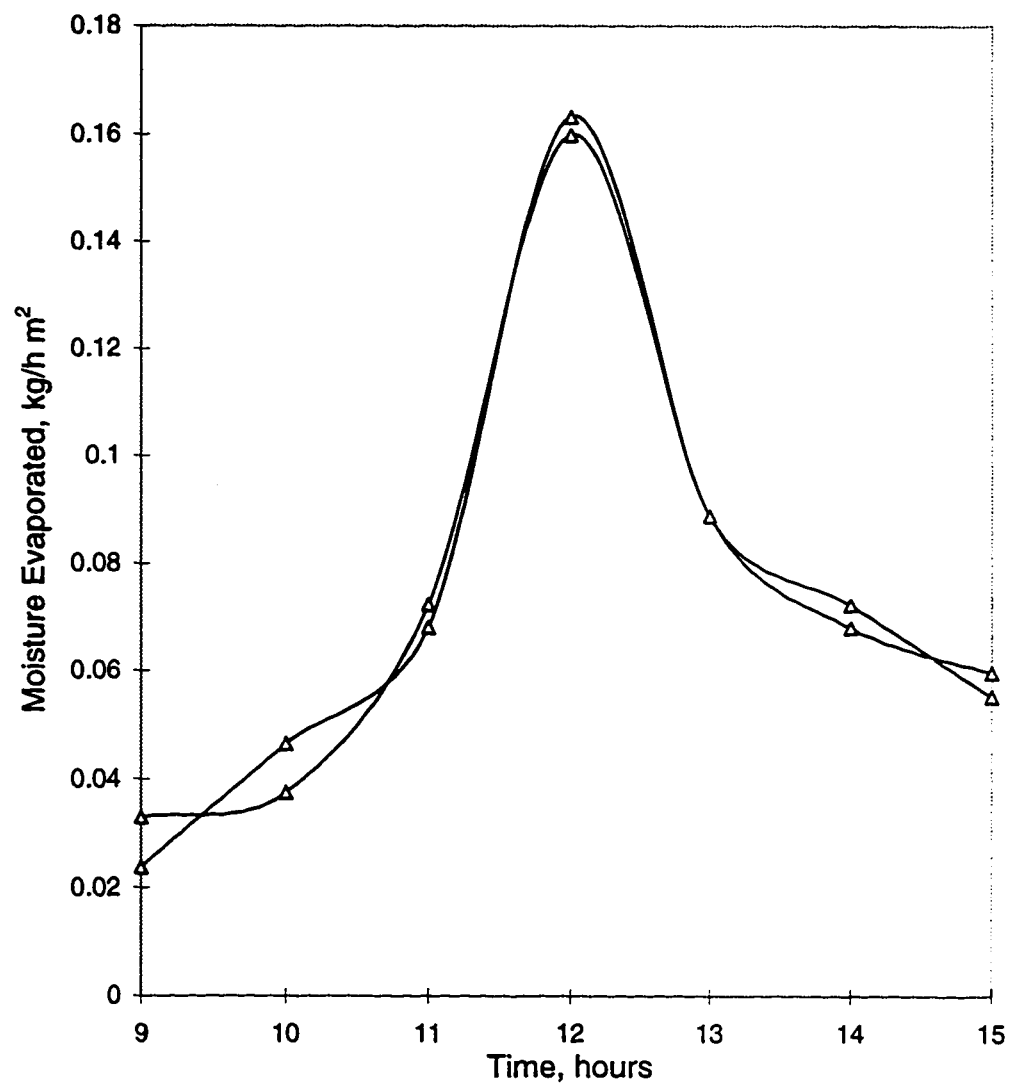


Figure 7.19: Moisture Evaporation Rate in a Fully Open Solar Regenerator.

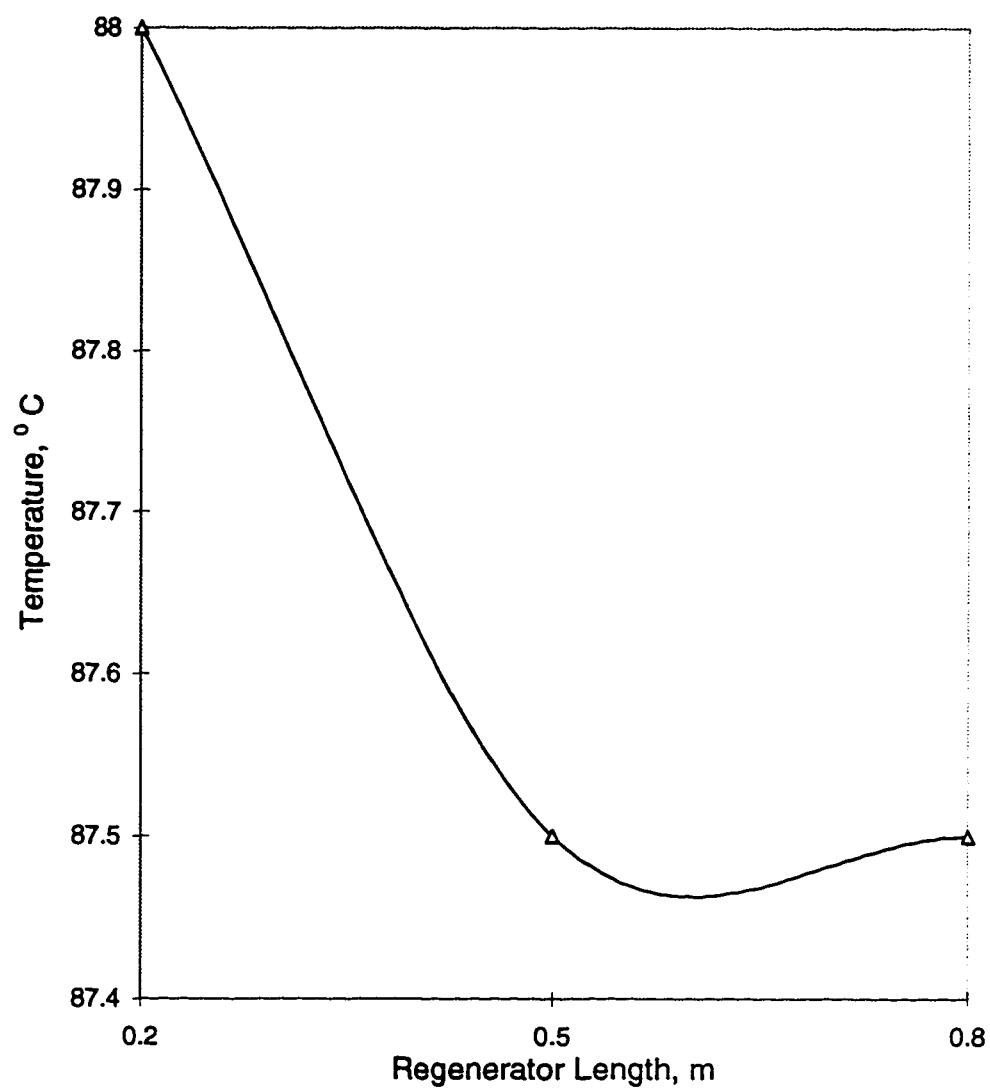


Figure 7.20: Temperature Profile in a Fully Closed Solar Regenerator.

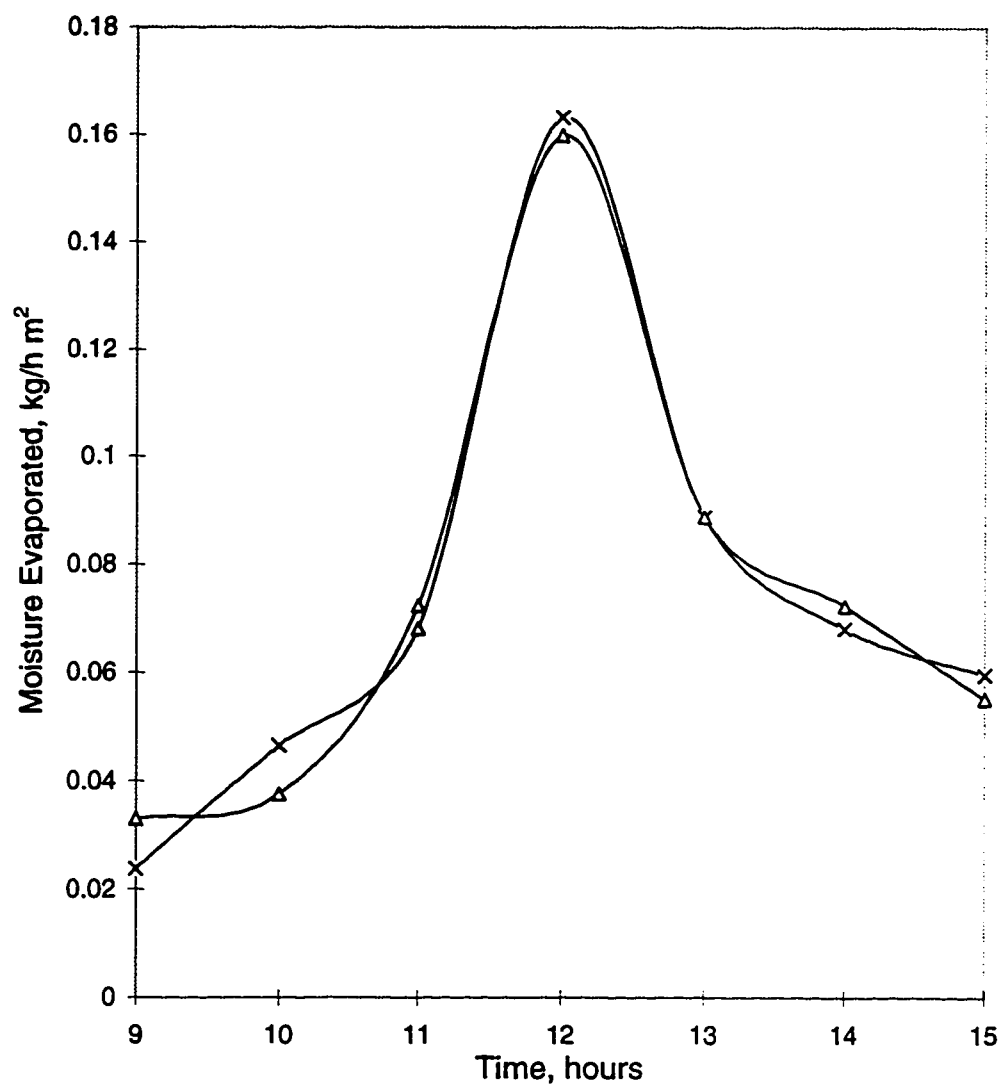


Figure 7.21: Moisture Evaporation Rate in a Fully Closed Solar Regenerator.

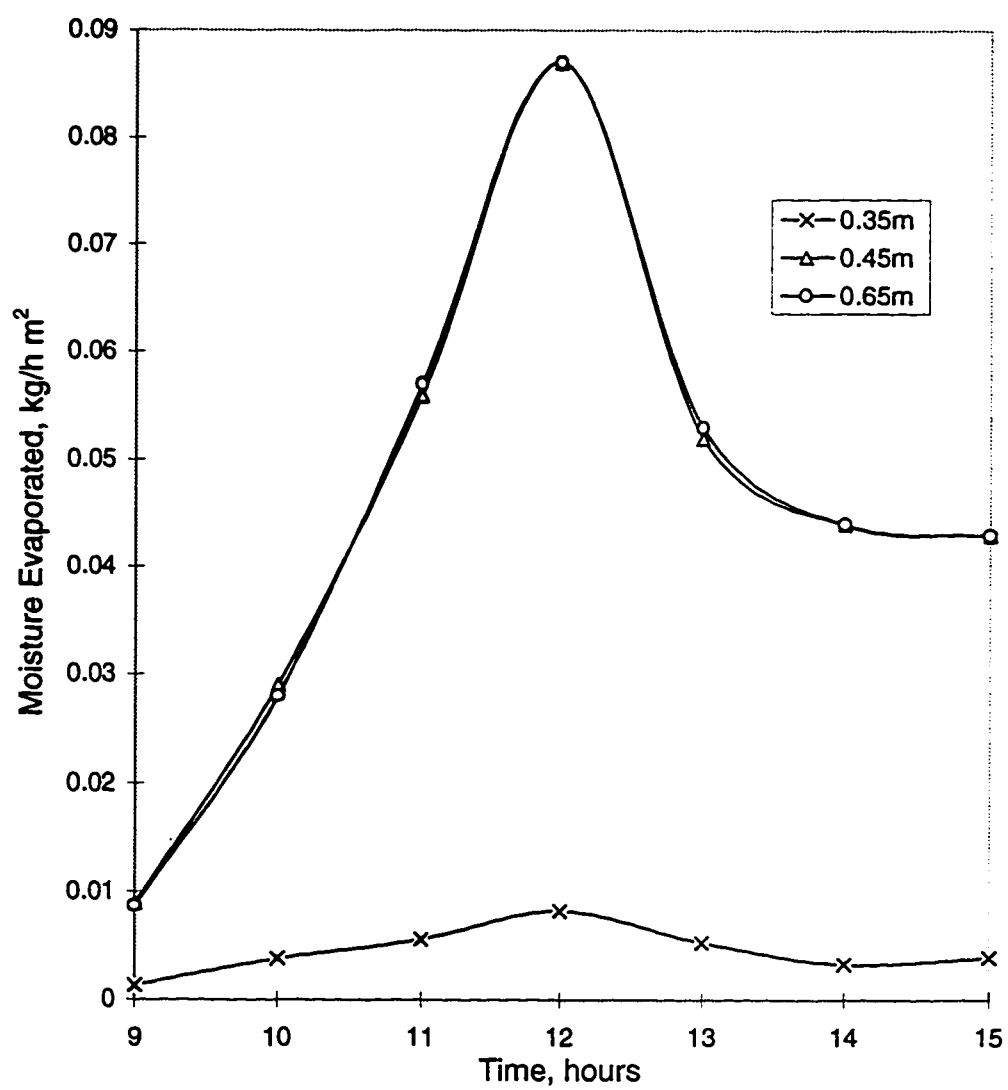


Figure 7.22: Moisture Evaporation as a Function of Glass Cover Height (Nov.1995).

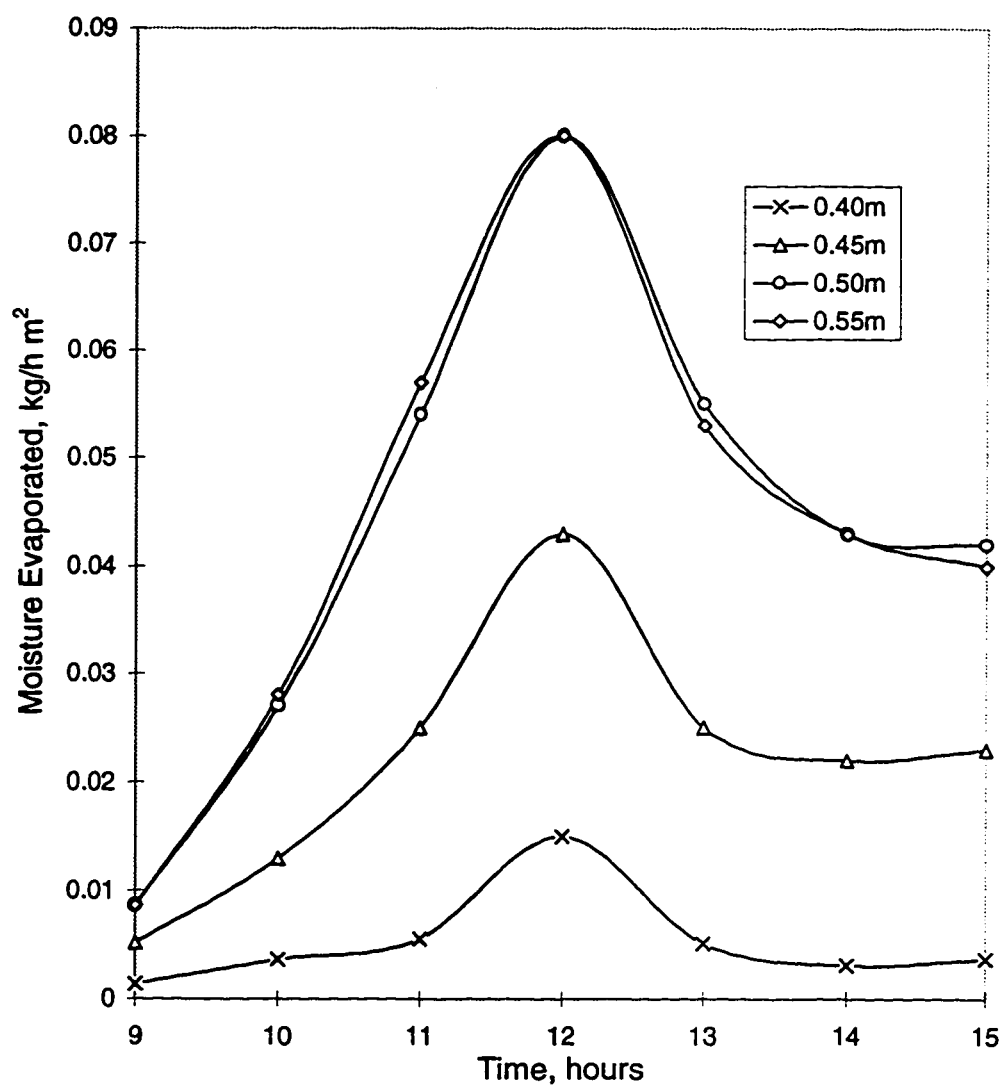


Figure 7.23: Moisture Evaporation as a Function of Glass Cover Height (June 1996).

regenerator was closed with a glass plate and part of the regenerator was kept open to the atmosphere. The glazed part was used to increase the temperature of the incoming desiccant solution and the open section was used to evaporate the moisture. The maximum moisture evaporation was found to be 0.116 kg/h m^2 . The variation of temperature along the regenerator plate for this type of regenerator is shown in Fig. 7.25. The variation of moisture evaporation along the time of the day is shown in Table 7.4. A maximum regenerator plate temperature was found to be 82°C . The temperature was found to decrease along the length of the regenerator plate in the open section, since the lower portion of the regenerator was open to the atmosphere. The glazed and open sections were of equal dimensions of $0.5\text{m} \times 0.5\text{m}$ each. The rate of moisture evaporation was found to be higher than that for the fully closed type of regenerator since the temperature of the desiccant solution was increased in the glazed section and the rate of moisture evaporation increased in the open section due to this increased temperature and the subsequent increase in desiccant vapor pressure.

The variation of moisture evaporation rate in alternately closed-open regenerator, along the time of the day, for Nov. 3 and 4 is shown in Fig. 7.26. In this type of regenerator, glass covers were used to cover the upper and lower portions of the regenerator while the middle portion was left open to atmosphere. That is, the regenerator was alternately closed and opened to the atmosphere. In every glazed section, temperature of the incoming desiccant solution was raised and the moisture

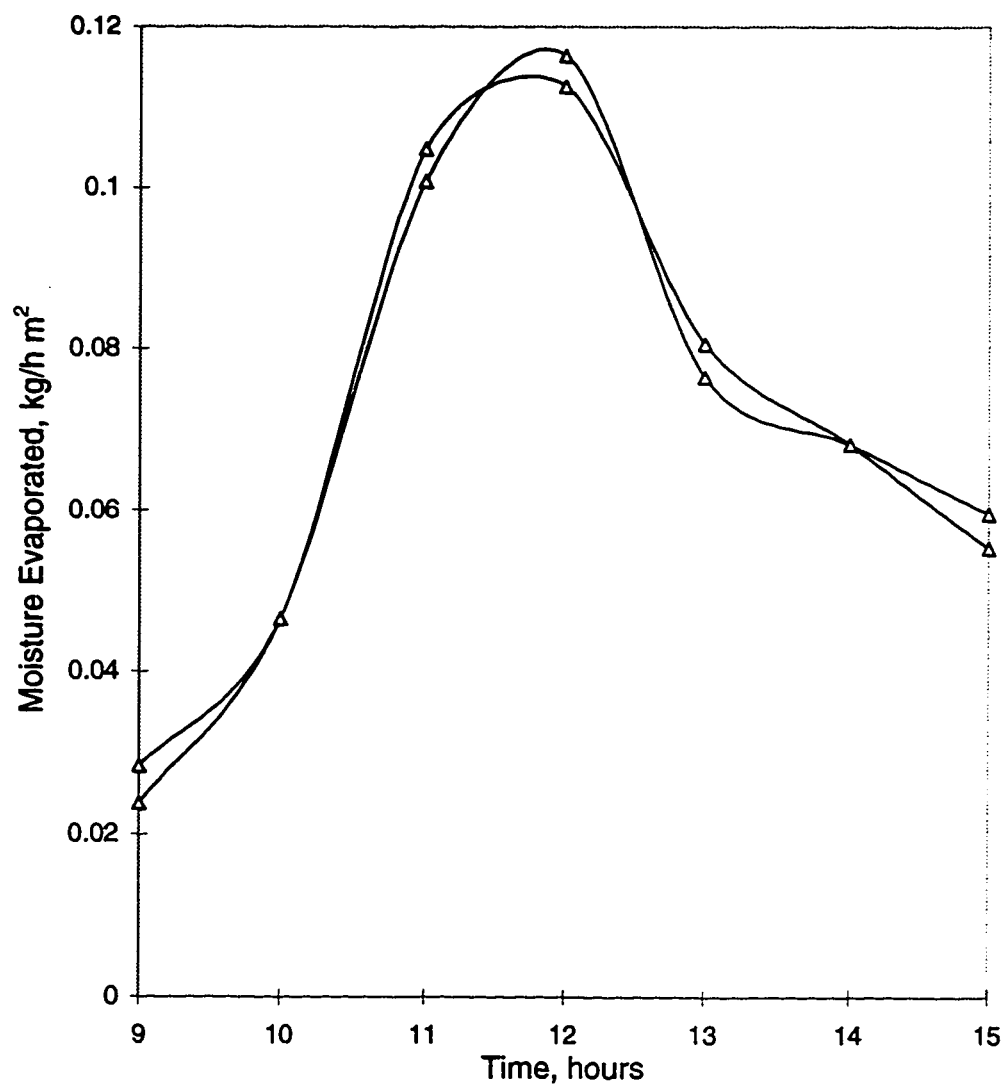


Figure 7.24: Moisture Evaporation Rate in a Partly Closed-Open Solar Regenerator.

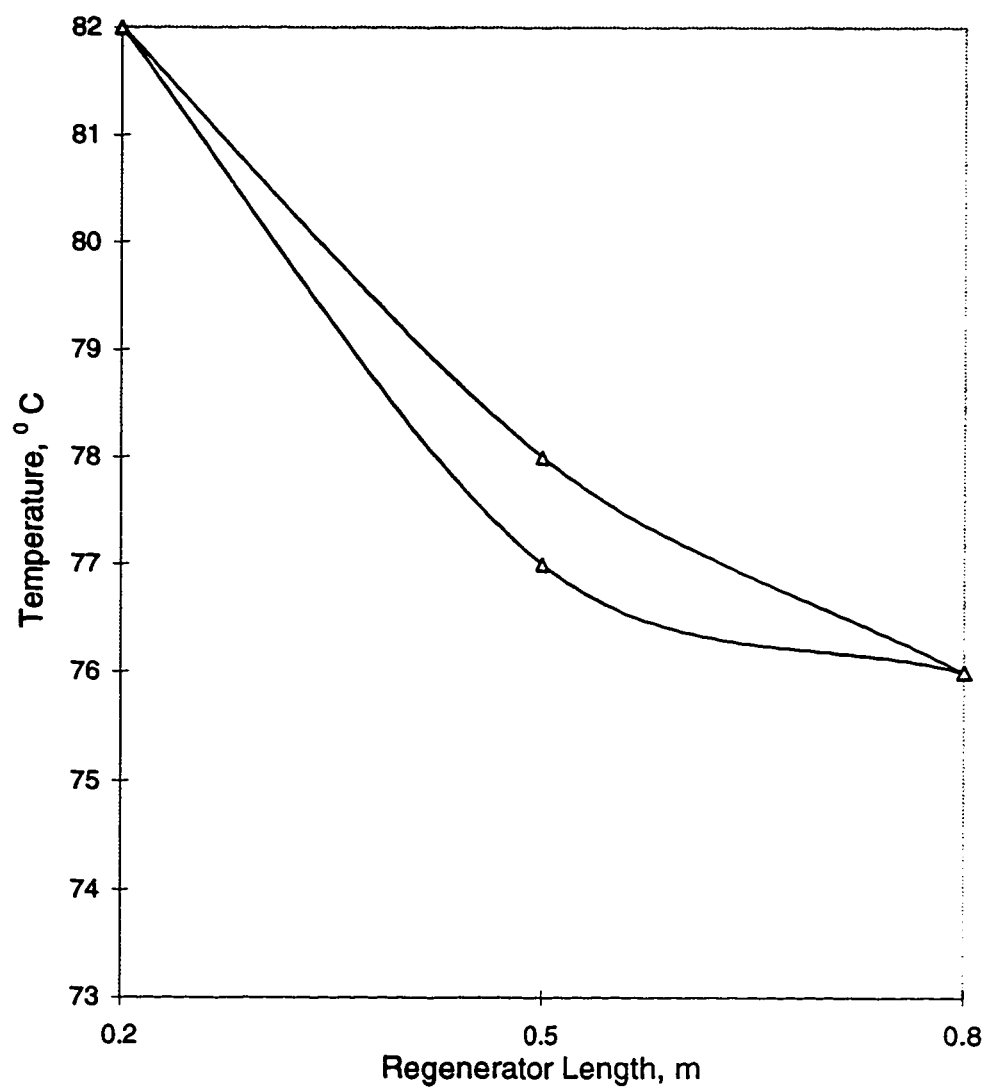


Figure 7.25: Temperature Profile in a Partly Close-Open Solar Regenerator.

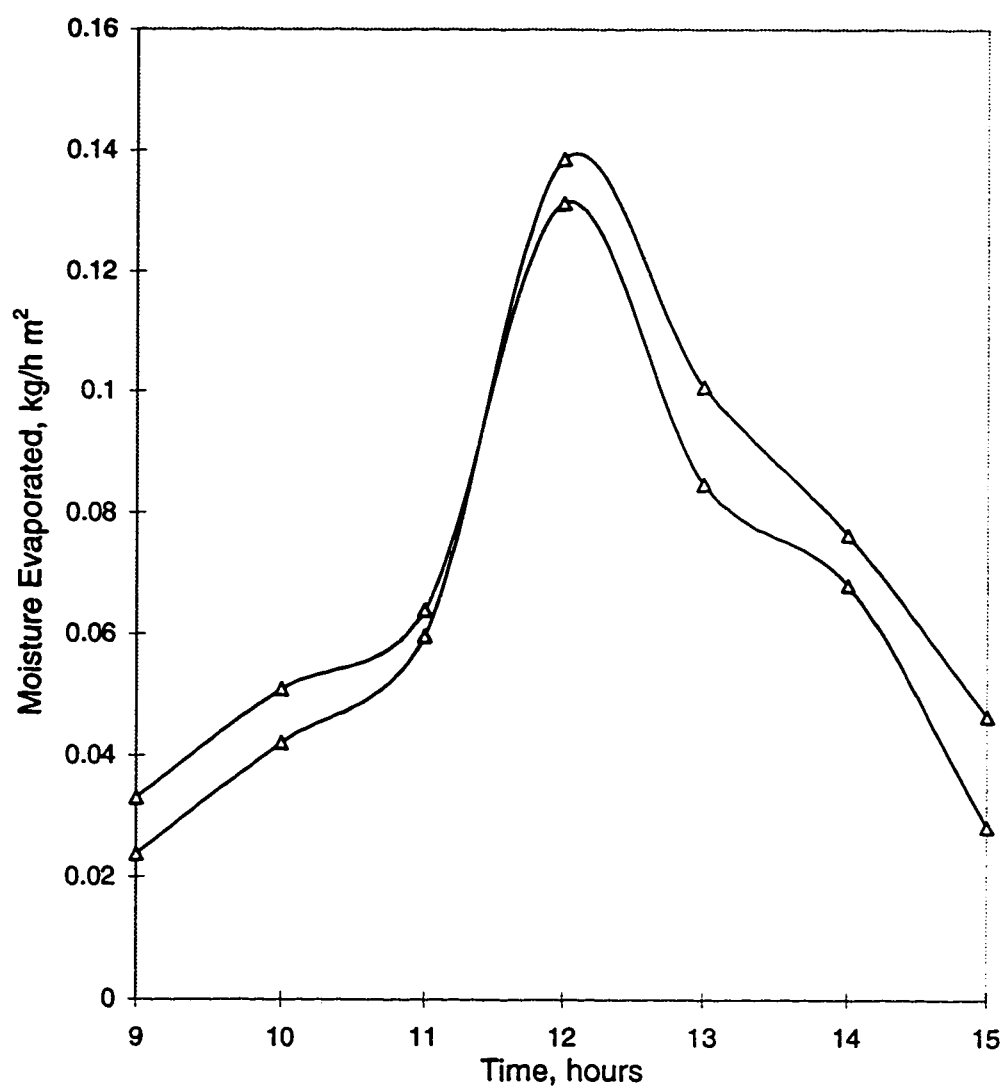


Figure 7.26: Moisture Evaporation Rate in an Alternately Closed-Open Solar Regenerator.

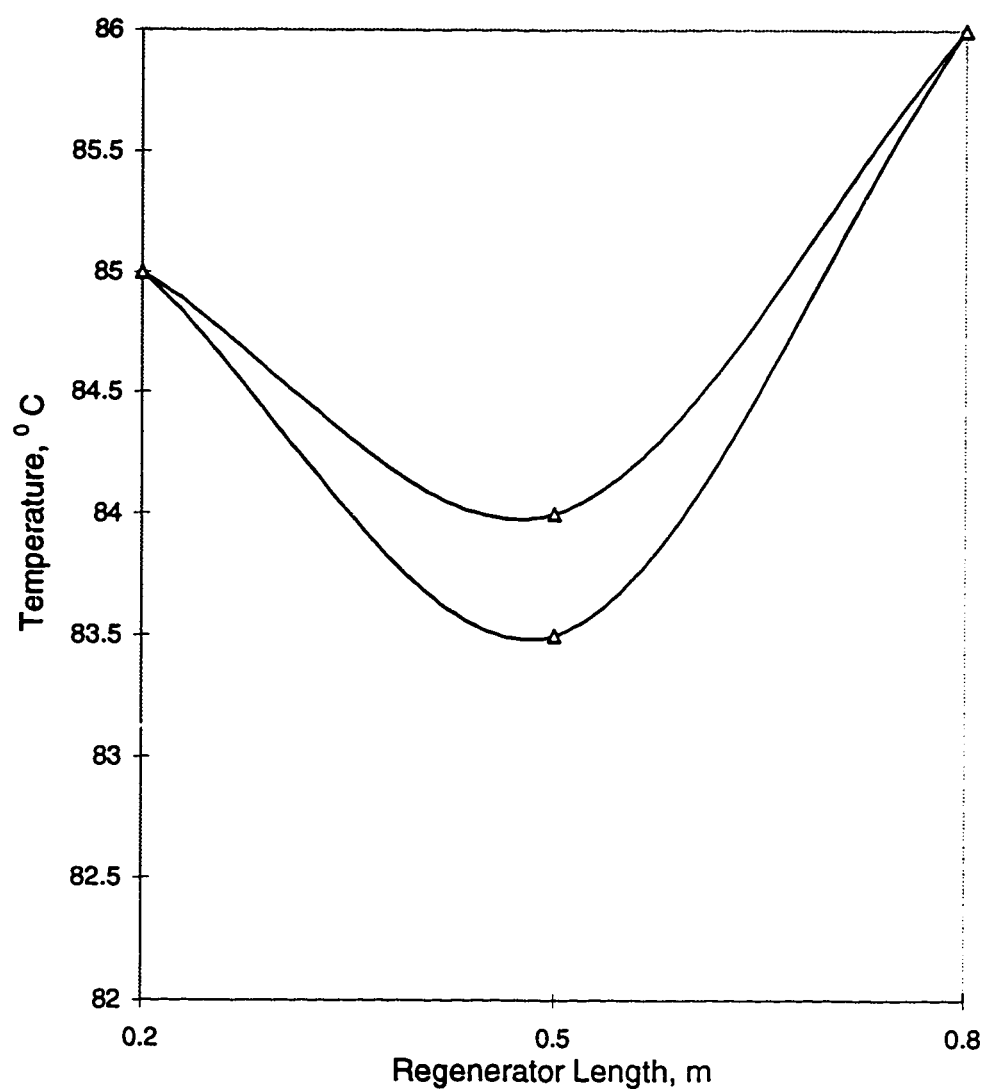


Figure 7.27: Temperature Profile in an Alternately Closed-Open Solar Regenerator.

evaporation took place in the open section. Thus, as the temperature was raised in the glazed section, the evaporation rate in the open section also increased. The temperature profile is shown in Fig. 7.27. The maximum temperature in this type of regenerator was found to be 85° C, while the maximum moisture evaporation was found to be 0.138 kg/h m² on 4 Oct. 1995. The dimensions of two glazed sections and one open section were 0.33mx 0.50m. The variation of moisture evaporation along the time of the day is shown in Table 7.5. The rate of moisture evaporation in this type of regenerator was found to be higher when compared to that of fully closed and partly closed-open, since in this regenerator two sections were glazed and one section was open. The desiccant surface temperature was also found to be higher in this type of regenerator. Hence it was found that the alternately closed-open regenerator performed better as far as moisture evaporation and desiccant temperature are concerned. But when compared to the fully open regenerator, this type of regenerator had less moisture evaporation rate.

Table 7.2: Experimental Results: Fully Open Solar Regenerator

Time, h	I,W-h/m ² (Oct.18)	I,W-h/m ² (Oct.19)	M, kg/h m ² (Oct.18)	M, kg/h m ² (Oct.19)
9	465	452	0.023	0.033
10	625	621	0.046	0.037
11	719	718	0.068	0.072
12	750	750	0.0163	0.015
13	711	713	0.088	0.088
14	618	608	0.068	0.072
15	456	441	0.059	0.055

Table 7.3: Experimental Results: Fully Closed Solar Regenerator

Time, h	I,W-h/m ² (Oct.25)	I,W-h/m ² (Oct.26)	M, kg/h m ² (Oct.25)	M, kg/h m ² (Oct.26)
9	430	420	0.014	0.009
10	583	580	0.037	0.02
11	678	690	0.059	0.050
12	716	726	0.088	0.088
13	636	684	0.059	0.050
14	525	577	0.037	0.042
15	401	424	0.037	0.042

Table 7.4: Experimental Results: Partly Closed-Open Solar Regenerator

Time, h	I,W-h/m ² (Oct.29)	I,W-h/m ² (Oct.30)	M, kg/h m ² (Oct.29)	M, kg/h m ² (Oct.30)
9	396	423	0.023	0.028
10	579	586	0.046	0.046
11	692	685	0.100	0.10
12	721	722	0.116	0.112
13	698	681	0.076	0.080
14	597	587	0.068	0.068
15	424	407	0.059	0.05

Table 7.5: Experimental Results: Alternately Closed-Open Solar Regenerator

Time, h	I,W-h/m ² (Nov.3)	I,W-h/m ² (Nov.4)	M, kg/h m ² (Nov.3)	M, kg/h m ² (Nov.4)
9	436	435	0.02	0.033
10	595	590	0.042	0.050
11	696	695	0.059	0.069
12	726	729	0.138	0.131
13	683	677	0.100	0.084
14	579	560	0.076	0.068
15	413	408	0.046	0.028

Chapter 8

CONCLUSIONS AND RECOMMENDATIONS

8.1 Conclusions

A comprehensive simulation model of a hybrid liquid desiccant and vapor absorption for the process of air-conditioning with a detailed heat and mass balance analysis for various components of the cycle has been presented. The analysis was made by coupling a liquid desiccant system for the process of dehumidification and a vapor absorption machine for sensible cooling. The model was developed to take into account the practical considerations in the variation of the parameters like absorber and evaporator temperatures, humidity ratios, and the ambient water (refrigerant) temperature. A maximum COP of 1.25 was obtained during the study and the unit

is found to be the best suited for hot and humid areas. For low ambient water (refrigerant) temperatures and humidity ratios the performance of the system is likely to be further increased.

The exergy analysis results indicate that exergy play a vital role in the performance evaluation of different components of the hybrid cycle. The regenerator has been given a special consideration since it is the major component of the hybrid cycle that consumes heat during the process of regeneration of the dilute solution. The optimum desiccant mass flow rate for minimum irreversibilities has been found to be about 30 kg/h m^2 for an ambient temperature of 40°C . Results also indicate that there is a considerable loss of exergy of the regenerator due to increase in the ambient vapor pressure. The maximum irreversibilities are generated at an ambient vapor pressure of 25 mm Hg and a desiccant mass flow rate of 5 kg/h m^2 . Hence the regenerator is expected to perform at its best at low ambient vapor pressures. Irreversibilities are also found to increase due to the increase in insolation and reduction in desiccant mass flow rate. Exergy analysis of the system components reveals that the irreversibilities in the absorber and the dehumidifier are almost equal since these two components have a similar function.

The rate of moisture evaporation in the alternately closed-open regenerator was found to be higher when compared to that of fully closed and partly closed-open type solar regenerators, since in this regenerator two sections were glazed and one section was open. The desiccant surface temperature was also found to be higher

in this type of regenerator. Hence, it was found that the alternately closed-open regenerator performed better as far as moisture evaporation and desiccant temperature are concerned. But when compared to the fully open regenerator, this type of regenerator had less moisture evaporation rate.

8.2 Recommendations

1. Dehumidification of air should be studied for different applications like moisture free metal coatings, better indoor air quality, pipe drying, etc.,.
2. The four different type of solar regenerators have to be fabricated and experimentally studied simultaneously for the performance evaluation. All the regenerators should be tested on the same day on which the parameters like insolation, ambient temperature, and ambient wind velocity are constant for all the regenerators.
3. Different combination of mixtures of two or more liquid desiccants should be tried in order to get better absorption performance without sacrificing the absorption properties and cost.

Appendix I

Liquid Water Enthalpy and Entropy

The enthalpy and entropy of liquid water are given by the following expressions [51]

Liquid Water Enthalpy

$$h_w = A + B + C - D + E - F + G$$

where

$$A=0.724252$$

$$B=3.81881T$$

$$C=0.0395435T^2$$

$$D=1.59977410^{-3}T^3$$

$$E=0.0296917110^{-3}T^4$$

$$F=0.25670610^{-6}T^5$$

$$G=0.838913110^{-9}T^6$$

Liquid Water Entropy

$$s_w = A + B - C + D - E + F - G$$

where

$$A=-1.87162310^{-3}$$

$$B=0.01631384T$$

$$C=0.123785810^{-3}T^2$$

$$D=3.83091610^{-6}T^3$$

$$E=6.99905910^{-6}T^4$$

$$F=0.608643710^{-9}T^5$$

$$G=2.00389910^{-12}T^6$$

Where T is in $^{\circ}\text{C}$, h_w is in J/kg, and s_w is in J/kg K

Water Vapor Specific Heat, Enthalpy, and Entropy

The specific heat, enthalpy and entropy of water vapor are given by the following expressions [51]

Water Vapor Specific Heat

$$c_v=14305-183.54\left(\frac{T}{100}\right)^{1/4}+82.751\left(\frac{T}{100}\right)^{1/2}-3.6989\left(\frac{T}{100}\right)$$

Water Vapor Enthalpy

$$h_v=2500800+1868T$$

and

Water Vapor Entropy

$$s_w = A - B + C - D + E - F + G$$

where

$$A=9.158667$$

$$B=0.02797079T$$

$$C=0.2511810^{-3}T^2$$

$$D=5.63091710^{-6}T^3$$

$$E=0.0978370410^{-6}T^4$$

$$F=0.0844698510^{-9}T^5$$

$$G=2.77680410^{-12}T^6$$

Where T is in $^{\circ}\text{C}$, h_v in J/kg , s_v in J/kg K and c_v is in kJ/kmole K .

Appendix II

Properties of Lithium Bromide and Water Solution

The relation for the concentration of the Lithium bromide in the solution is given by the expression [35]

$$X = \text{Exp}[((-P/32125) + 1.819)T - 16.42\text{LOG}(P) + 105.90]^{1/3} * 10$$

where P is in Pascals and T is in $^{\circ}\text{C}$

The relation for the enthalpy of Lithium bromide and water solution is given by,

$$H = A + B$$

where

$$A = A_0 + A_1X + A_2X^2 + A_3^3 + A_4X^4 + TB_0 + B_1X + B_2X^2$$

$$B = B_3X^3 + B_4X^4 + T^2C_0 + C_1X + C_2X^2 + C_3X^3 + C_4X^4$$

and

$$A_0 = -2024.33$$

$$A_1 = 163.309$$

$$A_2 = -4.88161$$

$$A_3 = 6.302948\text{E-}02$$

$$A_4 = -2.913705\text{E-}04$$

$$B_0 = 18.2829$$

$$B_1 = -1.1691757$$

$$B_2 = 3.248041\text{E-}02$$

$$B_3 = -4.034184\text{E-}04$$

$$B4 = 1.8520569E-06$$

$$C0 = -3.7008214E-02$$

$$C1 = 2.8877666E-03$$

$$C2 = -8.1313015E-05$$

$$C3 = 9.9116628E-07$$

$$C4 = -4.4441207E-09$$

Where T is in $^{\circ}$ C and X is in kg/kg.

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